SELECTED WATER RESOURCES ABSTRACTS
INPUT TRANSACTION FORM

1. Report No. R&D No 457

2.

3. Accession No. W

4. Title
Pumping and Energy Recovery Systems for Reverse Osmosis Desalination Plants

5. Report Date April, 1969

6.

8. Performing Organization Report No. SAL-5

7. Author(s) K. E. Hickman, J. P. Tramoni, J. T. Ganley, E. J. Fahimian

9. Organization
Dynatech R/D Co.
Cambridge, Massachusetts 02139

10. Project No.

11. Contract/Grant No. 14-01-0001-1462

13. Type of Report and Period Covered

12. Sponsoring Organization Office of Saline Water

15. Supplementary Notes

R & D #457

16. Abstract
This report covered a survey of pumping and energy recovery equipment for reverse osmosis desalination plants and identified areas in which further research and development work is desired in order to improve the operating characteristics and economics of the respective systems. Factors that influence water cost were analyzed.

17a. Descriptors

Desalination, reverse osmosis process

17b. Identifiers

pumping equipment, energy recovery equipment

17c. COWRR Field & Group

18. Availability

19. Security Class. (Report)

20. Security Class. (Page)

21. No. of Pages

22. Price

Send To:

WATER RESOURCES SCIENTIFIC INFORMATION CENTER
U.S. DEPARTMENT OF THE INTERIOR
WASHINGTON, D. C. 20240

Abstractor T. K. Lau

Institution Office of Saline Water, Membrane Division

BLANK PAGE

Research and Development Progress Report No. 457 • August 1969

Pumping and Energy Recovery Systems for Reverse Osmosis Desalination Plants

By K. E. Hickman, J. P. Tramoni, J. T. Ganley and E. J. Fahimian, Dynatech R/D Company, Div. of Dynatech Corp., Cambridge, Mass., for Office of Saline Water, W. F. Savage, Assistant Director, Engineering and Development; K. C. Channabasappa, Chief, Membrane Division.

Contract No. 14-01-0001-1462

UNITED STATES DEPARTMENT OF THE INTERIOR • Walter J. Hickel, Secretary
Carl L. Klein, Assistant Secretary for Water Quality and Research

As the Nation's principal conservation agency, the Department of the Interior has basic responsibilities for water, fish, wildlife, mineral, land, park, and recreational resources. Indian Territorial affairs are other major concerns of America's "Department of Natural Resources".

The Department works to assure the wisest choice in managing all our resources so each will make its full contribution to a better United States—now and in the future.

FOREWORD

This is one of a continuing series of reports designed to present accounts of progress in saline water conversion and the economics of its application. Such data are expected to contribute to the long-range development of economical processes applicable to low-cost demineralization of sea and other saline water.

Except for minor editing, the data herein are as contained in a report submitted by the contractor. The data and conclusions given in the report are essentially those of the contractor and are not necessarily endorsed by the Department of the Interior.

For sale by the Superintendent of Documents, U.S. Government Printing Office
Washington, D.C. 20402 - Price $2.00

TABLE OF CONTENTS

TABLE OF CONTENTS (Continued)

TABLE OF CONTENTS (Continued)

TABLE OF CONTENTS (Continued)

LIST OF ILLUSTRATIONS

LIST OF ILLUSTRATIONS (Continued)

LIST OF ILLUSTRATIONS (Continued)

LIST OF ILLUSTRATIONS (Continued)

LIST OF ILLUSTRATIONS (Continued)

LIST OF ILLUSTRATIONS (Continued)

LIST OF ILLUSTRATIONS (Continued)

LIST OF TABLES

LIST OF TABLES (Continued)

NOMENCLATURE

AC	Alternating Current
BEP	Best Efficiency Point
BHP	Brake Horsepower equal to: input HP for pumps output HP for turbines
¢/k gal	Cents per 1000 gallons
DC	Direct-Current (for electric motors)
DC	Direct Code Variable (computer programs)
DM	Deutsch Mark
EMF	Electromotive force
e_v	Volumetric efficiency (reciprocating pumps)
GPD	Gallons per day
gpm	Gallons per minute
H	Head (feet)
H_a	Absolute pressure at the free surface of the liquid (ft. of water)
h_e	Head loss in the suction pipe and impeller approach passages (ft. of water)
h_s	Static head of the free surface of the liquid above the pump center line (ft. of water)
h_v	Vapor pressure at the water temperature (ft. of water)
hp or HP	Horsepower
HP-hr	Horsepower-hour
KW	Kilowatts
KW-hr	Kilowatt-hour
M_b	Brine rejected flow rate (GPD)
$M_{F.W.}$	Fresh water production (GPD)
M_S	Feed water flow rate (GPD)
N	Rotational speed (rpm)
N_S	Specific speed
NPSH	Net Positive Suction Head (ft. of water)

P	Power (hp)
P_{atm}	Atmospheric pressure (equal to 15 psia)
P_{IN}	Membrane inlet operating pressure (psia)
P_{OUT}	Hydraulic turbine inlet pressure (psia)
PLTC	Plant power recovery system variable (computer program)
ppm	Parts per million (salinity level)
psia	Pounds per square inch absolute
psig	Pounds per square inch gage
Q	Capacity (gpm)
R	Recovery factor (%)
R	Gear speed ratio (Section 7.6)
rpm or RPM	Revolutions per minute
S	Suction specific speed
T	Torque
V	Volts
ΔP	Brine pressure drop (% of membrane inlet pressure)
η	Efficiency %
ρ	Average liquid density (equal to 64 lbm/ft^3)
σ	Thoma cavitation parameter

Manufacturers

B-J	Byron-Jackson
F-M	Fairbanks-Morse
I-R	Ingersoll-Rand
P-P	Peerless Pumps
Wx	Worthington

BLANK PAGE

SUMMARY

In the reverse osmosis process for water desalination, salt water pumped to pressures of 400 - 1500 psi is contacted with an osmotic membrane. The pure water which diffuses through the membrane constitutes the product. The remaining effluent brine is rejected.

A complete survey of existing means for high pressure pumping and energy recovery was carried out. Pumps and hydraulic turbines best suited for reverse osmosis desalination plants have been selected.

Information on performance, cost, availability and reliability of present equipment was specially prepared by the manufacturers for use in this study. Similar information was gathered for electric motors, steam turbines, and diesel engines for use as drivers for the pumps.

A significant portion of the cost of fresh water production is contributed by the capital costs, maintenance costs, and power costs for the high pressure pumping system. These "pumping system costs" are influenced by the type of pump selected, the pump speed, the type of driver and energy supply available, and the plant operating conditions. These include the salinity of the feedwater, feedwater flow rate, and membrane performance. This report shows the influence of these pumping system parameters upon fresh water cost for reverse osmosis desalination plants ranging in size from 10^5 GPD to 10^7 GPD of fresh water output. It was found that the initial cost of the equipment contributes only a small fraction of the total water cost. The principal contribution is made by the driver power cost, showing that emphasis should be placed upon using the most efficient equipment available.

A large fraction of the high-pressure water delivered to the membranes is rejected as concentrated brine. The economic advantage of using a hydraulic turbine to recover the energy in the high pressure brine was evaluated. It was found that energy recovery is not economically sound for 10^5 GPD plants. For 10^6 GPD and 10^7 GPD plants, energy recovery is economical.

Further development work appears to be desirable in several areas. Insufficient experience has been obtained by the pump manufacturers with respect to the selection of the best materials for saline water service in high pressure pumps and turbines. The operating life expectancy and maintenance requirements for pumps and turbines used in this service are not well defined. Because of the cost savings obtainable with high pump efficiencies, special centrifugal pump designs should be considered for the 10^5 and 10^6 GPD plant sizes. The standard pumps now available were designed for other purposes and consequently do not have the highest efficiencies attainable for the desalination plant application. New pump designs are needed to meet the requirements of the 10^7 GPD plant size. Present high-pressure pump product lines do not extend to such high flow rates.

Section 1

SCOPE OF INVESTIGATION

1.1 Introduction

In the reverse osmosis process for water desalination, salt water pumped to pressures of 400 - 1500 psi is contacted with an osmotic membrane. The pure water which diffuses through the membrane constitutes the product. The remaining effluent brine is rejected.

Due to the large flows of high pressure salt water needed, currently proposed systems require a large capital investment for the high pressure pump and its driver. Furthermore, since a large fraction of the high pressure water is rejected as concentrated brine, it may be economical to use a hydraulic turbine to recover part of the energy. Current knowledge of pumps and of hydraulic turbines suitable for applications in reverse osmosis plants is, however, incomplete. Data on the performance, cost, availability and reliability of these units is lacking. No work has been done to simplify the selection of the best type of pump and the most economical driving arrangement for particular plant conditions.

1.2 Purpose of This Study

This investigation had three objectives:

1. To conduct a state-of-the-art survey of standard pumping equipment that will meet the requirements for use in large-size reverse osmosis desalination plants. This survey includes a review of all of the types of pumps available for high pressure saline water service and identification of their principal operating characteristics. Also included is a review of standard driving systems for these pumps.

2. To determine the technical and economical feasibility of recovering energy from the high-pressure brine discharged by reverse osmosis desalination plants

3. To identify areas in which further research and development work is desirable in order to improve the operating characteristics and economics of pump and energy recovery systems for reverse osmosis desalination plants.

1.3 General Ground Rules

Table 1.3.1 shows the various elements that were included within the scope of this investigation and those elements which were not included. This table is not a complete list of all of the items that were and were not included in this survey. However, the major items of importance in the final economic analyses have been mentioned in this table.

INCLUDED	NOT INCLUDED
1) PLANT LAYOUT Pumping System (Pump & Driver) - Cellulose Membranes - Power Recovery System (hydr. turb. & gener.) - Gears	1) PLANT LAYOUT Piping System, Filters, Acid Tanks, Sumps, Circulating Pumps to Sumps, Valving, By-passes, Stand-by Units, Water Deaerators, Control Panels, Layout of Components in best locations, Pressure Vessels, Membrane Supporting Structure
2) PARAMETRIC STUDY Fresh Water Output, % Recovery (3 values) - Brackish W. Fresh Water Output, % Recovery (1 value) - Sea Water Feed Pressures (4 values) - Brine Pressure Drop (4 values) Average Density Value (64 lbm/ft^3) - Two Salinity Contents (1 brackish, 1 sea) - Acidity Level from Membrane Data - Membrane Flux, Average depending on Feed Pressure	2) PARAMETRIC STUDY Extra Pressure Losses due to Piping, Valving, Elbows, Controls, Flow-meters, Filters, Screens - Density variations due to Salinity Level, Pollution or Sand Contamination - Variable Fluxes Through Membranes (Time dependent)
3) PUMPING SYSTEMS Driver, Gear, Pump Combination - Reciprocating and Centrifugal Pumps - Cavitation, Materials, Aeration, Maintenance, Operation, Service Requirements	3) PUMPING SYSTEMS Positive Displacement Pumps (other than Recip.) - Detailed Analysis of Various Drivers (voltage, controls, overloads, etc.) - Geographical Location or Power Availability Considerations - Parallel Operation of Pumps.
4) ENERGY RECOVERY SYSTEMS Pelton impulse turbines, Governors, Electrical Generators, Gears - Maintenance, Operation, Service, etc.	4) ENERGY RECOVERY SYSTEMS Francis Turbines - Centrifugal Pumps as Turbines - Analysis of Controls and Regulation of Complete System
5) COST STUDY Component costs, Manufacturer Cost Correlations. Contributions to Water Cost - Optimization With and Without Energy Recovery - Influence of Output Rate, Feed Pressures, Losses, Types of Units - Average Cost Values used: Power, Membrane Costs (from OSW), Drivers, Gears, Pumps, Hydr. Turb., Maintenance, Service.	5) COST STUDY Cost Analysis for Specific Manufacturers and Pumps as a factor in Water Cost - Foundation, Installation, Delivery, Part Shipments, Repair Costs - Overhead Expenses, Plant General Costs, Administrative Costs - Structures and Improvements, Land Costs - Piping, Valving and Control Costs - Indirect Capital Costs - Water Treatment Costs (Sand and Bacteria) - No consideration of effects of location on Power Costs, Effect of Steam Supply Conditions, etc.
6) LITERATURE REVIEW & MANUFACTURER REVIEW	

Table 1.3.1 Ground Rules Followed For Pumping System Study

Section 2

GENERAL CHARACTERISTICS OF PUMPING AND ENERGY RECOVERY SYSTEMS FOR THE REVERSE OSMOSIS PROCESS

2.1 Introduction

A reverse osmosis desalination plant must operate continuously, delivering a steady flow of fresh water. The brackish or sea water entering the process must be raised to very high pressures before contacting the membranes. Thus, reliable high-pressure pumps are vital components of reverse osmosis desalination plants. The concentrated brine leaving the membranes remains at high pressure; consequently, energy recovery by means of hydraulic turbines may be economical. The purpose of this section is to review the types of hydraulic machinery suitable for pumping and energy recovery in the reverse osmosis process and to select the most promising types of equipment for a range of plant requirements. Table 2.1.1 summarizes the range of plant requirements and design parameters considered.

2.1.1 Pumping System Parameters

This study was intended to include a broad range of plant sizes, feed water salinity levels, and membrane performance characteristics. A parametric approach was used to meet these conditions. Several typical values each were selected for the following desalination process variables:

- plant size or fresh water output
- delivery pressure to the membranes
- fresh water recovery as a percentage of feed water flow

By studying the effects upon pumping system characteristics of all of the combinations of the selected values of these variables, the whole range of plant conditions could be covered effectively.

The plant sizes for the study were selected to be 100,000 GPD, 1,000,000 GPD, and 10,000,000 GPD of fresh water output independent of the salinity of the feed water.

Water Type	Design Operating Pressure: P_{IN} (psia)	% Recovery $R = \frac{M_{F.W.}}{M_s}$	Brine Pressure Drop $\Delta P = \frac{P_{IN} - P_{OUT}}{P_{IN} - 15}$ (%)	Plant Size: Output $M_{F.W.}$: GPD
Brackish (5,000 ppm)	400	50	0, 5, 10, 15	10^5, 10^6, 10^7
		70	0, 5, 10, 15	10^5, 10^6, 10^7
		80	0, 5, 10, 15	10^5, 10^6, 10^7
	600	50	0, 5, 10, 15	10^5, 10^6, 10^7
		70	0, 5, 10, 15	10^5, 10^6, 10^7
		80	0, 5, 10, 15	10^5, 10^6, 10^7
	800	50	0, 5, 10, 15	10^5, 10^6, 10^7
		70	0, 5, 10, 15	10^5, 10^6, 10^7
		80	0, 5, 10, 15	10^5, 10^6, 10^7
Sea (35,000 ppm)	1500	40	0, 5, 10, 15	10^5, 10^6, 10^7

Reverse Osmosis Desalination Plant
Design Parameters

For the purposes of this study, the water entering the process was characterized as either "brackish water" or "sea water". Brackish water was assumed to have a salt content of 5000 ppm, while sea water has a salt content of 35,000 ppm. The feed water must be delivered to the membranes at a pressure which depends upon its salinity. The delivery pressure levels assumed for this study were as follows:

- brackish water plants (5000 ppm) : 400, 600, or 800 psia
- sea water plants (35000 ppm) : 1500 psia

The proportion of the feed water which is converted to fresh water is a variable which depends upon salinity, delivery pressure level, amount of other impurities, and membrane characteristics. To allow for variations in the recovery percentage due to these factors, the following alternative values were used:

Brackish Water
- 50% of the feed water recovered as fresh water
- 70% of the feed water recovered as fresh water
- 80% of the feed water recovered as fresh water

Sea Water
- 40% of the feed water recovered as fresh water

From these assumptions it was possible to determine domains of operation for the pumping equipment and to define discrete pump sizes for the various plants.

The following equations relate the pump specifications to the plant conditions:

$$Q = \frac{M_{F.W.}}{R \times 24 \times 60} \quad \text{flow, gallons per minute}$$

$$H = \frac{(P_{IN} - 15) \times 144}{\rho} \quad \text{head, feet}$$

Figure 2.1.1 shows the limiting values for pump head and flow considered in this study. A rectangular domain represents the range of brackish water conditions while a straight line represents the range of sea water conditions.

2.1.2 Energy Recovery System Parameters

The fresh water recovered by the membranes is only a portion of the total incoming flow. The remainder of the flow, a concentrated brine at high pressure, can be directed through some type of hydraulic energy recovery device. The most suitable types of energy recovery systems for the selected plant conditions are hydraulic turbines and centrifugal pumps used as hydraulic turbines.

In order to determine the turbine inlet pressure corresponding to a given delivery pressure to the membranes, a parametric approach was used. Alternative values of pressure drop along the membrane channels of 0%, 5%, 10%, 15% were assumed.

The following equations relate the turbine specifications to the plant conditions and membrane pressure drop:

$$Q = (1 - R) \frac{M_{F.W.}}{R \times 24 \times 60} \quad \text{gallons per minute}$$

$$H = (1 - \Delta P) \frac{(P_{IN} - 15) \times 144}{\rho} \quad \text{head, feet}$$

Figure 2.1.2 shows the limiting values of head and flow available for energy recovery; one domain represents the range of brackish water conditions, the other represents the range of sea water conditions.

2.1.3 Membrane System Parameters

The performance of a reverse osmosis membrane varies with time as the material degrades. Both the operating pressure level and the cumulative time of operation influence the membrane flux rate. These combined effects are called membrane compaction. A typical set of curves of water flux variation with time in a continuous run is shown on Figure 2.1.3. The curves are steeper at higher pressure during the first hours of operation, then reach a fairly steady flux level. Staggered replacement of compacted membranes can be used to maintain a constant average flux rate and reduce operating cost.

Four values of average flux rates were provided by the Office of Saline Water for use in the study; these values depend upon the water salinity level and the pressure required for desalination as follows:

Liquid		Salinity		Pressure		Flux
Sea Water	-	35,000 ppm	-	1500 psia	-	10 GPD/ft^2
Brackish Water	-	5,000 ppm	-	800 psia	-	20 GPD/ft^2
				600 psia	-	15 GPD/ft^2
				400 psia	-	10 GPD/ft^2

The membrane area can be calculated approximately for a given plant size by using the following equation:

$$\text{Area} = \frac{\text{F. W. Output}}{\text{Water - Flux}} \quad (ft^2)$$

This area is the membrane area required to produce a specified amount of fresh water; it is a function of the delivery pressure level and the salinity of the feed water.

The recovery factor values indicated in Section 2.1.1 are additional characteristics of the membrane type. They are defined as follows:

$$\%\ \text{recovery} = R = \frac{\text{Fresh Water Flow Rate (output)}}{\text{Saline Water Flow Rate (input)}}$$

2.1.4 Plant System Definition

A schematic flow diagram of a reverse osmosis desalination plant is given in Figure 2.1.4.

The alternative plant design conditions described in the previous sections (2.1.1, 2.1.2, 2.1.3) have been summarized in Table 2.1.1. Using these plant design conditions, 30 pumping systems and a greater number of power recovery systems can be identified. The thirty pumping systems are described by a network relating flow rate and pressure, Figure 2.1.5. Table 2.1.2 presents the particular pump requirements (head and flow) corresponding to the 30 "standard" plants.

Head (feet)	Capacity (gallons per minute)		
867	#1 86.75	#4 867.5	#7 8,675
867	#2 99.30	#5 993	#8 9,930
867	#3 138.80	#6 1,388	#9 13,880
1320	#10 86.75	#13 867.5	#16 8,675
1320	#11 99.30	#14 993	#17 9,930
1320	#12 138.80	#15 1,388	#18 13,380
1765	#19 86.75	#22 867.5	#25 8,675
1765	#20 99.30	#23 993	#26 9,930
1765	#21 138.80	#24 1,388	#27 13,880
3340	#28 173.30	#29 1,733	#30 17,330

Table 2.1.2: Pump Design Requirements

We have submitted the values of head and flow listed in these tables to various manufacturers to obtain information on their products for use in the reverse osmosis process.

The information which was received in reply will be described in Sections 3 and 4.

2.2 Pumping Equipment

Positive displacement pumps and centrifugal pumps are standard equipment commercially available for producing a pressure rise in a flowing liquid. Positive displacement pumps operate by trapping a quantity of liquid and then forcing it out against an elevated discharge pressure. In centrifugal pumps the energy is imparted by centrifugal action: the liquid enters the pump near the axis of a high-speed rotating impeller and is thrown outward into the pump casing. The kinetic energy developed in the liquid by the vanes of the impeller is converted by the diffuser section of the casing into pressure head.

The various types of pumps can be characterized by domains of head and capacity where each type is most efficient. Figure 2.2.1 shows the domains of operation of positive displacement pumps and centrifugal pumps.

Section 2.2.1 presents a brief description of the different types of positive displacement pumps. Centrifugal pumps are described in Section 2.2.2. Those types of pumps which are not suitable for the reverse osmosis process are identified.

2.2.1 Positive Displacement Pumps

Positive displacement pumps can be classified under three main headings:

1. rotary pumps
2. reciprocating pumps
3. pumps that are a combination of types 1 and 2.

Rotary pumps transfer the liquid from suction to discharge by means of rotating gears, lobes, vanes or screws operating inside a rigid container. Figure 2.2.2 presents sectional views of various types of rotary pumps:

Internal and External Gear Pumps - The liquid is trapped and moved by the gears as indicated on the cross-section. The liquids pumped must be free of solids to avoid erosion, and free of gases to avoid cavitation problems. The liquid also must be adequate as a lubricant for the gears. Gear pumps will have wearing problems when operated with sea water.

Lobe Pumps - Operate on the same principle as gear pumps but tend to wear less than gear pumps. The output from lobe pumps pulsates more than that from gear pumps.

Vane Pumps - Consist of an eccentric rotor and rectangular vanes that can slide radially. As the rotor revolves, the vanes are forced out against the fixed casing by centrifugal force. The liquid is moved from suction to discharge in the space between the rotor and the fixed casing. Pumping rates are changed by varying the rotor speed as well as its eccentricity. These pumps are self-priming and produce constant, uniform discharge flow rate. Wearing problems are confined principally to the vanes which are self-compensating until worn out. Seals and relief valves are required, and foreign bodies can damage the pump.

Screw Pumps - These pumps can be of two types: single screw or twin screw. In a single screw machine, a helical screw rotor revolves in a shaped stator. In a twin screw machine, the two screws rotate in opposite directions. In either case, the liquid is caught in a cavity which progresses towards the discharge end of the pump. The discharge pressure dictates the length and pitch of the helical screw rotor. These pumps are self-priming and very reliable. Liquids containing vapors, gases and solids can be pumped. Screw pumps cannot be operated against a closed discharge; relief valves and seals are needed. They are heavy and bulky and are very sensitive to variations in discharge pressure.

Flexible Impeller Pumps - These are eccentric-rotor pumps. The rotor is equipped with flexible blades that are bent against a fixed casing. The blades unfold when passing the suction part and draw in the liquid. When the blades are bent, they squeeze the liquid and force it into the discharge port of the pump. These pumps are self-priming, operate with all sorts of liquids, and have a constant, uniform discharge flow rate. They can be serviced easily. However, they have a limited pressure range and are not suitable for heavy-duty applications.

Reciprocating Pumps - Includes those machines which provide energy to the liquid through the reciprocating cyclic action of a piston or a plunger in a cylinder. The output flow rate of these units varies sinusoidally with time. The discharge flow fluctuations can be reduced by use of several pistons operating in parallel. Figure 2.2.3 shows various types of reciprocating pumps.

There are other types of positive displacement pumps such as diaphragm pumps (where a flexible diaphragm replaces the piston), eccentric-cam pumps (either constant-volume, or variable-volume) and peristaltic pumps (a flexible tube is squeezed by rollers at each end of a rotor and the liquid is pushed to the discharge). These pumps are shown in Figure 2.2.4. All of these pumps are self-priming and do not require shaft seals or check-valves. They can move the liquid in either direction without bringing it into contact with the moving parts. Diaphragm pumps can reach high pressures but the others are quite limited in both capacity and pressure.

Figures 2.2.5 and 2.2.6 both describe the ranges covered by these positive displacement pumps:

- maximum attainable discharge pressure
- maximum attainable capacity

2.2.2 Centrifugal Pumps

A great variety of centrifugal pumps have been built for various applications. The pumping of liquids or generation of head is accomplished by a rotary motion of one or several impellers. On the basis of the main direction of discharge of the liquid, centrifugal pump impellers are classified as follows:

- Radial flow

- Mixed flow

Every pump consists of two principal parts, an impeller which forces liquid into a rotary motion by impelling action, and the pump casing which directs the liquid to the impeller and leads it away under high pressure.

Centrifugal pumps can also be broadly classified on the basis of internal casing design into the following types:

- Volute

- Diffuser

- Turbine

Figures 2.2.7 shows cross-sections of these different designs. In a volute casing, the impeller discharges into a single channel of gradually increasing area called a volute, and the major part of the conversion of kinetic energy to pressure head takes place in the conical discharge nozzle. In a diffuser casing, the major part of the conversion of velocity into pressure takes place between the diffuser vanes. In a turbine pump, the liquid does not discharge freely from the tip of the impeller but is recirculated back to lower points on the impeller diameter where it recirculates many times before finally leaving the impeller. These pumps develop high heads. The impeller vanes rotate in an annular channel in the pump casing. The casing contains a sealing wall through which the impeller passes with very close clearances.

There are many other means of characterizing centrifugal pumps:

- impeller shape and operating characteristics

- enclosed, semi-enclosed, or open impellers

- single or double-suction impellers

- external casing design (vertically or horizontally split-case, or barrel-type case)

- vertical or horizontal rotating shaft
- single or multistage pump
- position of the pump in relation to the liquid supply (wet- or dry-pit mounted, or in-line)

2.2.3 Preliminary Selection of Pumps for Reverse Osmosis Plants

The previous sections have described many different types of pumps. Some of these pumps will not match the requirements of the desalination plants considered in this investigation. Factors to be taken into account in the selection of a pump are as follows:

1. Duty cycle, operating conditions
2. Operating speed
3. Liquid used
4. System cleanliness
5. Efficiency

Based upon these factors, the choice of pumps must be restricted to reciprocating pumps and multistage centrifugal pumps (volute or diffuser casing, horizontal or vertical shafts). The following pumps have been eliminated from further consideration due to factors (1) and (3):

- Gear Pumps: (internal and external) water is non-lubricating and these pumps have a pulsating output which is undesirable for the membranes.
- Lobe Pumps: do not meet pressure and flow requirements
- Vane Pumps: continuous high pressure duty would cause failure
- Screw Pumps: heavy and bulky units, are not standard for the conditions required.

- Flexible Impeller Pumps: cannot meet the high pressure requirements.

Only two types of pumps appear to be suitable for the desalination plant requirements for this study:

- Reciprocating pumps for high pressure and low capacity requirements

- Centrifugal multistage pumps for all other applications.

Factors (2) and (4), speed and system cleanliness, are not decisive in the final selection of pumps, since drivers and gears are available at all speeds and the feed water is to be treated before entering the pumping system. Factor (5), the pump efficiency, will be used to accomplish a second selection, on the basis of power savings, between centrifugal units and reciprocating units for use at the lower capacities.

2.2.4 Pump Performance

a. Performance Curves and System Curves

The performance characteristics of a pump are generally described at a particular rotational speed by two curves, the head vs. capacity curve and the efficiency vs. capacity curve. The flow rate vs. input pressure level for the flow system connected to the pump can be described by a third curve. Figure 2.2.8 shows these three curves. The intersection of the "system" curve and the pump head-capacity curve is the operating point of the pump.

At constant speed, a reciprocating pump delivers essentially the same capacity at any pressure within the power capability of the driver and the maximum pressure limitation of the pump. Reciprocating pumps are highly efficient units. Due to the reciprocating action of the piston or plunger, the flow-rate varies cyclically around a mean value. Figure 2.2.9 describes their flow variations for different types of reciprocating pumps. These theoretical flow curves are indicative of the pressure pulsation which can be expected in the operation of reciprocating pumps.

The performance of a centrifugal pump is described by its rate of flow or capacity, Q, and its head, H, in feet of the liquid pumped. Other performance characteristics are the power input and the speed of rotation. Four types of head-capacity curves for constant speed operation may be identified (Figure 2.2.10). The steepness of the curve varies (a, b, c): the maximum head is developed at zero capacity and the head decreases as the flow rate is increased. The slope of the head-capacity curve is dictated by the geometry of the impeller and pump casing. Curve (d), a "drooping curve" leads to unsteady operation. Operating at only fixed head above the zero capacity value, the pump oscillates between two different capacities.

Figure 2.2.10 also displays three types of power vs. capacity curves:

a. is referred to as non-overloading

b. describes a normal overloading curve where power increases with capacity

c. illustrates an overloading curve where power increases with a decrease in capacity.

The use of two pumps working together in a pumping system introduces additional constraints on the operation of each pump. If the pumps are operated in parallel, the total head of each pump must be the same and equal to the total head for the system. If the pumps are operated in series, the total capacity of each pump must be the same as the total capacity of the system.

b. Pump Efficiency and Specific Speed

In reciprocating pumps a distinction is made between three efficiencies:

– The volumetric efficiency, e_v, which is the ratio of the actual liquid volume discharged to the piston or plunger displacement volume. Volumetric efficiency is sometimes replaced by the slip ($S = 1 - e_v$), the volumetric flow loss as a percentage of displacement. This efficiency parameter is reduced by leakage past the piston packing, the stuffing-box packing and the valves. The compressibility factor of liquids must be taken into account in the calculation of the volumetric efficiency.

– The hydraulic efficiency accounts for the hydraulic loss due to liquid friction in the cylinder and the pressure drop through the valves.

– The mechanical efficiency, which includes the previous hydraulic efficiency, is the water horsepower delivered by the pump divided by the horsepower input to the pump. This efficiency provides a measure of the mechanical friction in the bearings and seals of the pump together with the hydraulic losses within the pump.

Reciprocating pumps are very efficient units. Their mechanical efficiencies are of the order of 85% to 90% and their volumetric efficiencies usually reach 98% or more.

The efficiency of a centrifugal pump is defined as the ratio of the water horsepower delivered by the pump to the horsepower required to drive the pump.

$$\eta = \frac{\text{Water horsepower}}{\text{Input horsepower}} = \frac{(hp)_1}{(hp)_2}$$

$$(hp)_1 = \frac{\text{gpm x feet}}{3960}$$

$$(hp)_2 = \text{horsepower provided by driver (BHP)}$$

A typical efficiency vs. capacity curve for a centrifugal pump is shown on Figure 2.2.10. The efficiency of a centrifugal pump depends upon its specific speed (hydraulic design), capacity, internal running clearances, surface roughnesses in the impeller and casing, and stuffing-box friction.

Figure 2.2.11 shows the efficiencies of single-stage pumps as a combined function of both specific speed and capacity. The specific speed, N_S, is an index that characterizes the operating conditions for a pump.

$$N_S = \frac{N\ (\text{rpm}) \times \sqrt{Q\ (\text{gpm})}}{(H)^{3/4}\ (\text{ft})}$$

The specific speed parameter also is used in several other ways:

- It is a fundamental, dimensionless parameter used to describe the performance of a pump at its most efficient operating point. As such, values of N, Q and H leading to the same value of N_S for geometrically similar pumps also lead to similar flow conditions within these pumps.

- For any value of specific speed, there is one type of pump which is more efficient than all other types of pumps at that specific speed. Thus, specific speed can be used as a criterion for selecting the most efficient type of pump for a particular operating condition. A reciprocating pump is categorized as a low specific speed pump, while axial flow pumps are high specific speed pumps.

At a given rotational speed, a high-head, low-flow impeller will have a low specific speed, large diameter, and narrow passages. A low-head, high-flow impeller will have a high specific speed, small diameter, and large passages.

The efficiency of multistage centrifugal pumps takes into account the losses due to the passages between the different stages as well as the individual stage efficiencies.

If the stuffing-box pressure is unnecessarily high, the efficiency of a pump will be decreased since extra power will be needed to overcome the increased friction.

Figure 2.2.11 shows that low specific speed pumps are less efficient than high specific speed pumps. In low specific speed pumps, friction losses are high in the narrow flow channels and disk friction losses and leakage losses are larger in proportion to the power input.

c. Cavitation

Cavitation is the formation of vapor bubbles in a flowing liquid. In a pump, cavitation occurs if the local vapor pressure at some point in the pump inlet flow falls below the vapor pressure. When the cavities or bubbles are conveyed by the flow into regions of higher pressure, they collapse. The stress waves resulting from the collapse can damage adjacent surfaces of the pump.

The pressure level at any point in the pump inlet flow is influenced by the following:

- the lift of the pump (the height of the first impeller inlet above the liquid free surface)
- pressure losses due to friction in the system upstream of the pump and between the pump inlet and the point in question
- atmospheric pressure (altitude)
- local flow velocity, including streamline curvature effects

The vapor pressure in the liquid is set by its temperature.

Cavitation causes noise and vibration as well as possible structural damage. However, it is difficult to employ the noise as a possible sign of cavitation; if a pump is operated off-design, the noise generated by the pump as a whole will mask the cavitational noise. A more reliable criterion for cavitation detection is a drop in pump efficiency and also in the head-capacity performance curve.

Cavitation damage takes the form of pitting which always occurs beyond the low pressure points in the pump inlet (Figure 2.2.12). Cavitation pitting can be due to stress pulses repeated at high frequencies. Metal particles are torn off and carried away by the liquid penetrating into and escaping from the pores of the metal under the influence of the intense pressure waves. This phenomenon is often described as "corrosion fatigue".

To avoid cavitation, the liquid must be admitted in the suction port of the pump at a pressure above a certain level called the "MINIMUM NET POSITIVE SUCTION HEAD".

$$NPSH = H_a + h_s - h_v - h_e$$

where:

H_a: absolute pressure at the free surface of the liquid. This is atmospheric pressure if the suction vessel is open to atmosphere, or the absolute pressure of the gas in this vessel if it is enclosed.

h_s: static head of the free surface of the liquid above the pump center line. *If it is suction lift, this head becomes negative*

h_v: vapor pressure at the water temperature

h_e: head loss in the suction pipe and impeller approach passages

Two dimensionless parameters are used to define cavitation limits for geometrically similar pumps:

- the suction specific speed S
- the Thoma cavitation parameter, σ

$$\sigma = \frac{NPSH}{H} = \frac{\text{Net Positive Suction Head}}{\text{Total Head}}$$

$$S = \sigma^{-3/4} \cdot N_S$$

The cavitation parameters σ and S remain constant for the same pump at different speeds, or for similar pumps at the same specific speed, if the pumps are operated at conditions satisfying the affinity laws.

Material selection is an important factor influencing the life of pump components exposed to cavitating flows. Figure 2.2.13 shows test results on material resistance to cavitation pitting. Material selection is not the only factor affecting cavitation damage: a small amount of air in the liquid is beneficial to reduce or prevent cavitation pitting. Air bubbles in small amounts have a cushioning effect and damp the stress waves caused by the implosions of vapor pockets.

d. Priming

Before any pump is started it must be fully primed; that is, the casing and suction pipe must be completely filled with the liquid to be pumped. Entrance of air into the pump or suction line during operation will break the prime and the pump will have to be reprimed.

Reciprocating pumps are self-priming. Centrifugal pumps usually need to be primed unless they are submerged. Some centrifugal pumps are self-priming and are able to exhaust the air from the suction pipe and the impeller. These pumps are of two types:

- Pumps equipped with a "suction chamber" having a suction nozzle higher than the impeller. These pumps are self-priming only if there is no discharge back pressure (open discharge).

- Pumps equipped with an air pump to prime the main pump. The air pump runs in parallel with the water pump and exhausts the air from the suction duct. Two types of air pumps are used, the so-called "side-canal" type (impeller with radial ribs) and the "water ring" type (an impeller which is eccentrically positioned inside the casing). These air pumps are usually equipped with two-way, automatic valves at their discharge nozzle. While priming, the valve remains opened. When water is pumped by the air pumps, the valve closes the air outlet and returns the water to the suction (Figure 2.2.14).

Other priming systems can be used for centrifugal pumps:

- flooded suction (positive head on the suction)
- ejectors or exhausters operated by compressed air, steam or water against a closed discharge
- vacuum pump (preferably a wet vacuum pump)
- automatic priming devices
- use of a reciprocating or positive displacement pump

2.3 Driving System

2.3.1 Power Supply

The driver selected for a pump will depend upon the various kinds of power supplies available and the costs of these power supplies. The three types of power supplies considered in this study are:

- electricity
- high-pressure steam
- diesel fuel

The relative costs and availability of these fuels will depend to a great extent on the location of the desalination plant and the quantity of power required.

Several types of equipment could be used as drivers. These types will be discussed briefly in the sections below. A choice between the types will depend on the factors:

- availability
- reliability
- capital cost

- maintenance cost
- space requirements
- starting and shutdown conditions
- speed variation requirements
- installation problems
- cooling problems
- air pollution problems

Before a final selection is made, all of these factors must be weighed.

2.3.2 Electric Motors

Motor operation is based on the principle that a conductor carrying current in a magnetic field tends to move in a direction perpendicular to the field. The conductors of a motor rotate relative to the magnetic field and are driven by the field. At the same time, the conductors generate an electromotive force (EMF) by generator action. This induced EMF is in opposition to the terminal voltage and tends to oppose the flow of current entering the armature.

There are three basic types of electric motors:

- direct-current motors
- synchronous motors
- induction motors

Direct Current Motors - There are several types of direct-current motors.

- Shunt motor: In this particular type, the flux is substantially constant. The armature and the field are in parallel. Hence, the speed varies only slightly with load so that the motor is suitable for service requiring constant speed.

- Series motor: In the series motor, the armature and the field are in series. Therefore, if saturation is neglected, the flux is proportional to the current and the torque varies as the square of the current. Thus, any increasing current will produce a much greater increase in torque, so this motor is suitable for service requiring large starting torques. However, the speed of series motors is practically inversely proportional to the current. Unsafe speeds may be reached if the load is allowed to drop off completely.

Other types of direct-current motors are the differential compound motor (with an adjustable speed) and the cumulative compound motor (very large starting torques).

The speed of direct-current motors may be controlled quite easily. Speed control may be achieved in several ways without altering the motor construction:

- vary the armature EMF: the armature of the motor is connected across different voltages. The control is accomplished by having power supplies which are maintained at different voltages available for the motor.

- field control: this method, which is used for speed control of series motors, is achieved by inserting a resistance in series with the motor. This method has the disadvantages of low efficiency and poor speed regulation for fluctuating loads.

- armature resistance control: this method is used for shunt motor speed control. An external resistor is inserted in the armature circuit only. This method is simple to accomplish and introduces no commutating difficulties. This control method allows development of the full torque of the motor at any speed at the expense of low efficiency and poor speed regulation with fluctuating loads.

Synchronous Motors - These motors have the unique property that their speed does not change as the applied load varies from no load to maximum load. For starting, the synchronous motor is connected to an AC power supply through a high resistance to produce high torque. This resistance is cut out as the speed increases. As synchronism is approached, the rotor windings are connected to a DC power source and the motor operates synchronously. This class of motors also includes slip-ring motors which are synchronous units equipped with phase-wound dampers connected to external resistors through slip-rings.

Induction Motors - Various types of induction motors are available.

- Polyphase induction motor: This is the most common type of electric motor. Its stator is wound in the same manner as the synchronous generator stator. There are two types of rotors: the squirrel-cage type consisting of heavy copper bars short-circuited by end-rings, and the wound-rotor having a polyphase winding with the same number of poles as the stator, and terminals brought out through slip-rings so that external resistance may be added.

- Other types include the double-squirrel-cage motor, the single-phase induction motor and the alternating current commutator motor.

Electric motors are widely used as drivers for pumps.

Cumulative compound DC motors are commonly used to drive single-acting reciprocating pumps, while multiplex pumps can be driven with shunt motors. For efficient operation, the method of speed regulation by field control is generally used for DC motors.

Shunt motors, differential compound motors, synchronous and squirrel-cage induction motors are used as drivers for centrifugal pumps. These motors satisfy the typical centrifugal pump requirements of small starting torque and high operating speeds.

Of these types, squirrel-cage induction motors are generally recommended by pump manufacturers as preferred drivers for reciprocating pumps or centrifugal pumps.

The required motor output power rating is set by the power needed to drive the pump at its maximum operating conditions, accounting for the gear and coupling efficiency. Induction motors are usually supplied with a service factor of 1.15. Thus, these motors can operate continuously at 115% of their rated power without causing any harm to the motor or its insulation.

When the reverse osmosis plant contains an energy recovery system, the motor rating might be affected. When a hydraulic turbine is used to recover energy from the high pressure waste brine, the turbine can be coupled directly to the pump or can be coupled to an electrical generator which supplies energy to the electric motor. In the latter case, the motor must be rated to drive the pump at its maximum operating conditions. During startup of the plant (before the generator is set into motion), the motor will be drawing as much as 50 percent more electrical power from the outside supply than during normal generator operation. The amount of power required from the outside supply will decrease as the water pressure builds up and the generator is set into motion

In a plant designed with the hydraulic turbine connected directly to the pump. the motor power during full plant operation is the difference between the total power required by the pump and the power recovered by the turbine. During plant startup, however, the turbine will lag behind the rest of the system in its power output, and will be supplying full recovered power only after the pump has reached its full head level. For this reason, there will be a short period during which the motor will experience a greater load than during normal operation. Therefore, the motor power rating should be slightly higher than that required during full load operation.

For this study it will be assumed that the usual 1.15 power factor is sufficient to accommodate the higher starting load.

2.3.3 Steam Turbines

The steam turbine is a very flexible type of driver. Turbines can be designed to run on almost any steam condition. They are used to drive many different types of machines such as electric generators, pumps, and compressors. When designed for variable speed operation, a turbine can be run efficiently over a considerable range of speeds; this is an important advantage in many applications. Steam turbines range in output capacity from a few horsepower to as much as 500,000 horsepower.

Turbines are classified in various ways:

1. By steam supply and exhaust conditions: i.e., condensing, non-condensing, automatic extraction, mixed pressure (in which steam is supplied from more than one source at more than one pressure), regenerative extraction, and reheat.

2. By casing or shaft arrangement; single casing, tandem compound (two or more casings with a shaft,coupled together in line), cross compound (two or more shafts not in line, often at different speeds).

3. By number of exhaust stages having parallel steam flow, e.g., double flow and triple flow.

4. By details of stage design; impulse or reaction.

5. By direction of steam flow in the turbine; axial flow, radial flow, tangential flow.

6. Whether single-stage or multistage.

7. By type of driven apparatus, e.g., generator drive or mechanical drive.

Any particular turbine unit may be described using one or more of the classifications.

Compared with other prime movers, steam turbines require less floor space, lighter foundations, and less attendance by an operator. No internal lubrication is required for the flow-handling components; hence the exhaust steam is free from oil. They have no reciprocating masses with their resulting vibrations. The discharge flow is uniform and steady. Turbines have no rubbing parts. They have a great overload capacity, high reliability, low maintenance costs, and excellent speed regulation. In purchasing a steam turbine, one must specify the inlet steam conditions (temperature and pressure), the discharge pressure, the power to be delivered and the shaft speed.

For a turbine designed to be most efficient at a given set of conditions, the efficiency (horsepower per pound of steam) will vary as the shaft speed changes. Turbine efficiency and investment costs increase with higher inlet steam pressures and temperatures and lower exhaust pressures. The steam consumption can be decreased by super-heating because, for a fixed set of inlet and outlet pressures, the energy available is proportional to the absolute inlet steam temperature.

Figure 2.3.1 shows the variation of turbine efficiency as function of turbine rating in kilowatts when the inlet steam conditions vary. Increases in inlet temperature above 1200° F must await the development of higher strength materials of reasonable cost and availability.

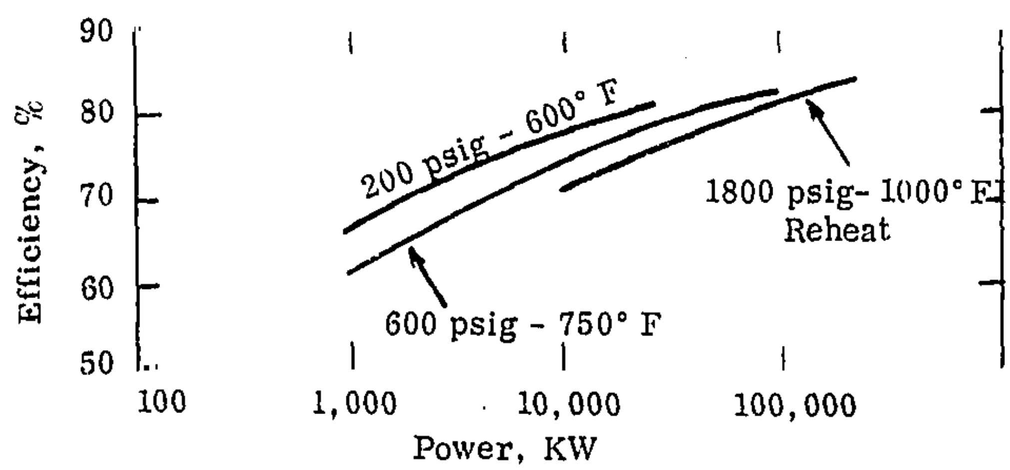


Figure 2.3.1 Steam Turbine Efficiencies Vs. Power
For Various Inlet Steam Conditions
(From Ref. 117)

Steam turbines can be adapted to the steam requirements of the plant:

- "Straight-through" types have a single high pressure inlet and single low pressure outlet.

- "Bleeder" turbines contain one or more intermediate outlets to provide intermediate pressure steam for plant process requirements.

- Non-condensing turbines have exhaust pressures from 50 to 100 psia.

- Condensing turbines are supplied with a condenser to condense the exhaust steam to water at a low pressure.

A non-condensing steam turbine is used when steam at a relatively high pressure and temperature can be purchased and returned, sold, or used within the plant at a reduced temperature and pressure. Because the steam is not reduced to a very low pressure, the turbine consumes a large amount of steam per horsepower delivered and the operating cost is high. On the other hand, because the steam is not condensed, the capital cost of a condenser is not required and the expense of maintaining a supply of cooling water is not present. Furthermore, because the exhaust is not at a very low pressure, the steam turbine is smaller in size and less expensive.

A condensing steam turbine can use inlet steam having the same temperature and pressure as a non-condensing unit. The exhaust is condensed, however, in a condenser utilizing cooling water. The condensate is generally pumped back at high pressure to the boiler to be reheated. Because of the low exhaust temperature available with a condenser, the steam can be expanded to a much lower pressure than is possible in a non-condensing unit. Therefore, the turbine utilizes a greater proportion of the energy of the steam. A well-designed condensing steam turbine should be able to operate with only 50 percent of the steam required by a non-condensing turbine having the same power output. The disadvantages of the condensing turbine are: its higher initial cost, the cost of the condenser, and the cost of maintaining a cooling water supply. The chief differences between the two types of turbine operation are summarized in Table 2.3.1

Steam Turbine Type	Capital Cost	Steam Cost	Typical Inlet Steam Conditions	Typical Exhaust Conditions
Condensing	High	Assumed Identical (OSW data)	200 psig - 600° F to 600 psig - 750° F	50 psig - Saturated From condenser: Water
Non-condensing	Low			

Table 2. 3. 1 Comparison Between Condensing and Non-Condensing Steam Turbines

2. 3. 4 Diesel Engines

The diesel engine is one of the most economical types of internal combustion engines. Diesel engines are available in sizes from a fraction of a horsepower to as large as 20,000 horsepower. They are designed to operate at speeds of 2,000 rpm or lower; a very common speed is 720 rpm. The diesel engine is superior to steam turbines or electric motors with respect to power-to-weight ratio, and external power supplies (electricity or steam) are unnecessary.

In order to utilize a diesel engine as a driver for the pumps considered in the present study, it is necessary to use gears between the engine and the pump. A stronger shaft will be needed on the pump because of the high levels of torsional vibrations which are transmitted from the engine.

The type of fuel consumed by diesel engines is readily available throughout the United States at a cheaper cost than electricity and steam.

However, capital costs and maintenance costs of diesel engines are substantially higher than those of steam turbines and electric motors. For this reason, diesel engines are not widely used to drive pumps.

2. 4 Energy Recovery System

2. 4. 1 Energy Recovery Potentials

When a liquid must be throttled from a higher pressure to a lower pressure, a potential for energy recovery exists. In reverse osmosis desalination plants, large flow rates of concentrated brine are discharged at high pressures from the membrane channels. Energy recovery can be achieved from this brine by using hydraulic turbines or centrifugal pumps operated as turbines.

The energy recovered from the waste brine becomes available as shaft horsepower and can be utilized as follows:

- to provide part of the shaft power required to drive the pump
- to drive an electrical generator tied into the plant power supply or producing electrical power for sale

The economic advantage of energy recovery depends upon the brine flow rate and pressure, the efficiency of the hydraulic machine, its capital and operating costs, and its reliability.

2.4.2 Hydraulic Machines for Energy Recovery

Hydraulic turbines and centrifugal pumps used as turbines can be used to transform potential energy in high-pressure liquids to shaft power. There are three types of hydraulic turbines:

- the impulse type, or Pelton wheel turbine, suitable for high heads.
- the reaction type, or Francis turbine, suitable for medium heads
- the propeller type, or Kaplan turbine, for low head applications

All of these turbines have a stationary casing with guiding passages (nozzles) in which the static head is transformed partly, or wholly, into velocity. These passages discharge into a runner. The impulse turbine guide passages transform the head into velocity. In reaction turbines, this transformation is only partial, and there is an additional pressure drop in the runner.

Figure 2.4.1 shows a cross-section of each of these types of turbines. In impulse turbines a nozzle directs a jet of water into the buckets of the runner. In reaction turbines the water enters the wheel radially and leaves the shrouded buckets in the axial direction to discharge into a draft tube. The power output of reaction turbines is controlled by wicket gates.

Impulse turbines usually are mounted with a horizontal shaft, while reaction turbines often have a vertical shaft.

Figure 2.4.2 shows the domains of efficient operation of these machines. The flow handled by the machine dictates its size.

The speed of a turbine is limited by the mechanical strength of the runner, by vibration and cavitation considerations, and if connected to a generator, by the synchronous speeds required for AC power generation. The speed should be as high as permissible since the turbine and generator will then be less expensive.

If the turbine rotor is allowed to revolve without load and the wicket gates (or needle) are wide open, it will approach its "runaway speed". Thus, speed controls are necessary. Furthermore, any overspeed requirement will appreciably increase the cost of the generator.

Hydraulic turbines are very efficient machines. Their efficiencies vary from 85% up to 94%.

The specific speed, N_S, is an important dimensionless parameter for hydraulic turbines as well as pumps, since it is related to the maximum head and cavitation limits.

$$N_S = \frac{N P^{1/2}}{H^{5/4}} = \frac{N Q^{1/2}}{H^{3/4}}$$

N is the rotational speed in rpm

P is the power in hp

H is the head in feet

Q is the capacity in gpm

Figure 2.4.3 shows the specific speed versus head curve for reaction and Pelton turbines. The other plot shows the "Thoma cavitation parameter", σ, versus specific speed for Francis and Kaplan turbines.

The critical cavitation condition must not be exceeded for Francis or propeller Kaplan turbines; otherwise, frequent shutdowns for inspection and maintenance of eroded surfaces will be required. The cavitation condition is not a critical factor for Pelton wheels since atmospheric pressure exits at the bucket discharge. However, Pelton wheels are subject to wear due to the impingement of the high-velocity jet.

Hydraulic turbines are not available as standard lines of equipment. They must be designed according to the specific conditions of flow and head available. When standard equipment is needed, centrifugal pumps are often used as turbines. The flow in a centrifugal pump used as a turbine is reversed by applying the head to the discharge nozzle. A good centrifugal pump can be used as a reasonably efficient hydraulic turbine. However, a pump used as a turbine will be less efficient than in its normal operation.

Type	Specific Speed N_S	Efficiency as a Pump	Efficiency as a Turbine
Radial Flow	1800	83%	70%
Mixed Flow	7500	82%	78%
Axial Flow	13500	80%	78%

Figure 2.4.4 is a Karman-Knapp diagram that shows the different zones of operation of a radial-flow centrifugal pump.

2.4.3 Preliminary Selection of Hydraulic Turbines

In order to recover the maximum amount of shaft power available in the waste brine, the most efficient hydraulic machines should be used. This consideration already eliminates the centrifugal pumps from further investigation since their efficiency will always be lower than that of a properly-designed hydraulic turbine.

The head levels available in most of the "standard plants" selected for this investigation suggest the use of Pelton impulse wheels. In plants with heads below 1000 feet and flow rates of 10^7 gallons per day, Francis turbines also can be used.

For turbines used in systems where the liquid under pressure is not obtained from a natural fall, normal practice recommends the use of Pelton impulse wheels. Pelton impulse wheels are smaller in size and do not require a draft tube for discharged flow. They are very reliable and easy to service.

2.4.4 Electric Generators

An electric generator is a rotating machine which transforms mechanical energy into electrical energy. When the armature is rotated through a transverse magnetic field, a voltage is induced in the conductive armature coil. When a resistor is connected across the rotating coil, a force is exerted on the conductor in a direction to oppose the rotation of the armature. The addition of a resistor results in the generation of electrical energy and the consumption of mechanical energy supplied to the shaft.

The rating of an electrical generator is based upon temperature limitations of the various parts of the machine. The nameplate specifies the rating in terms of:

- electrical or mechanical output
- voltage
- speed and overspeed limitations
- current
- temperature rise above ambient.

"Full load" is the rated power output of the machine. Electrical generators, like electric motors, are very efficient machines.

2.4.5 Power Transmission Devices

Couplings are required to transmit power from the driver (electric motor, steam turbine or diesel engine) to the pump, and from the hydraulic turbine to either an electrical generator or the pump. A number of different types of power transmission couplings are available. The choice of the best type to use must be based upon considerations ot the particular components of the plant and their arrangements. Possible types of power couplings are:

a. Direct mechanical shaft couplings

b. Belts and chains

c. Gears

d. Electric couplings

e. Hydraulic couplings

When power is to be transmitted at constant speed from the driver to the pump, both can be mounted in-line and coupled with a simple flanged coupling which directly connects the shafts. If it is not possible to place the components in-line, but it is feasible for their shafts to be placed in parallel, belts and chains or idler gears could be used. Belts and chains are not considered most practical for high power and high speed and are restricted to use in the smaller plants.

When it is desired to transmit power from one shaft to another and at the same time effect a speed reduction or increase, belts or chains can be used for the smaller plants, and gears can be used for higher speeds and large plants. For cases in which the machinery shafts are not parallel, gears are the most practical type of mechanical couplings

The type of gears to be used in each of the applications depends upon the power level, speed, and arrangement of the components within the plant. Spur gears are most economical for power levels lower than 100 HP. External helical or herringbone gears are preferred for larger power levels. Bevel gears can be used to couple shafts at right angles, but they are expensive and seldom used in pumping system applications. Parallel or coaxial shaft arrangements are strongly preferred.

Electric and hydraulic couplings are generally used with machinery with unusual starting characteristics, and wh. .e it is desired to vary the torque of the two components independently.

2.5 Unconventional Systems

A study of unconventional schemes for pumping water and recovering energy from the rejected concentrated brine was planned as one phase of the project. Some of the concepts considered were as follows:

- Condensing Steam Ejectors as Pumps:

  Condensing steam ejectors can pump feedwater to the pressure levels required in the reverse osmosis process. However, their efficiency is very low (of the order of 5%) so they are not practical unless waste heat is readily available.

- Rotating Membrane Devices:

  Advances in membrane technology might permit the membranes to be mounted at the circumference of rotating cylinders. Fresh water would be forced through the membrane by centrifugal action while the rejected brine can flow out along the axis at low pressure. This arrangement combines pump, membrane, and hydraulic turbine action in one single rotating unit.

None of these concepts, or others which were considered, proved to be attractive at this time. For the plant sizes considered in this study, the equipment efficiency is the principal factor controlling the pumping system's contribution to fresh water cost. The initial cost of the pumping and energy recovery equipment is a secondary factor. It is difficult to propose new pumping schemes or energy recovery schemes which have the potential of being more efficient than conventional pumps and turbines.

The most attractive methods for reducing the pumping system's contribution to the cost of fresh water appear to be in the area of total energy conservation. The complete energy balance of a plant, including the source of energy (steam generator , electrical source, or fuel oil), the driver, the pump, the hydraulic turbine and its form of energy output, should be considered as a system optimization problem. This problem must be solved for each particular plant because its location, size, and output demand variations will influence the solution.

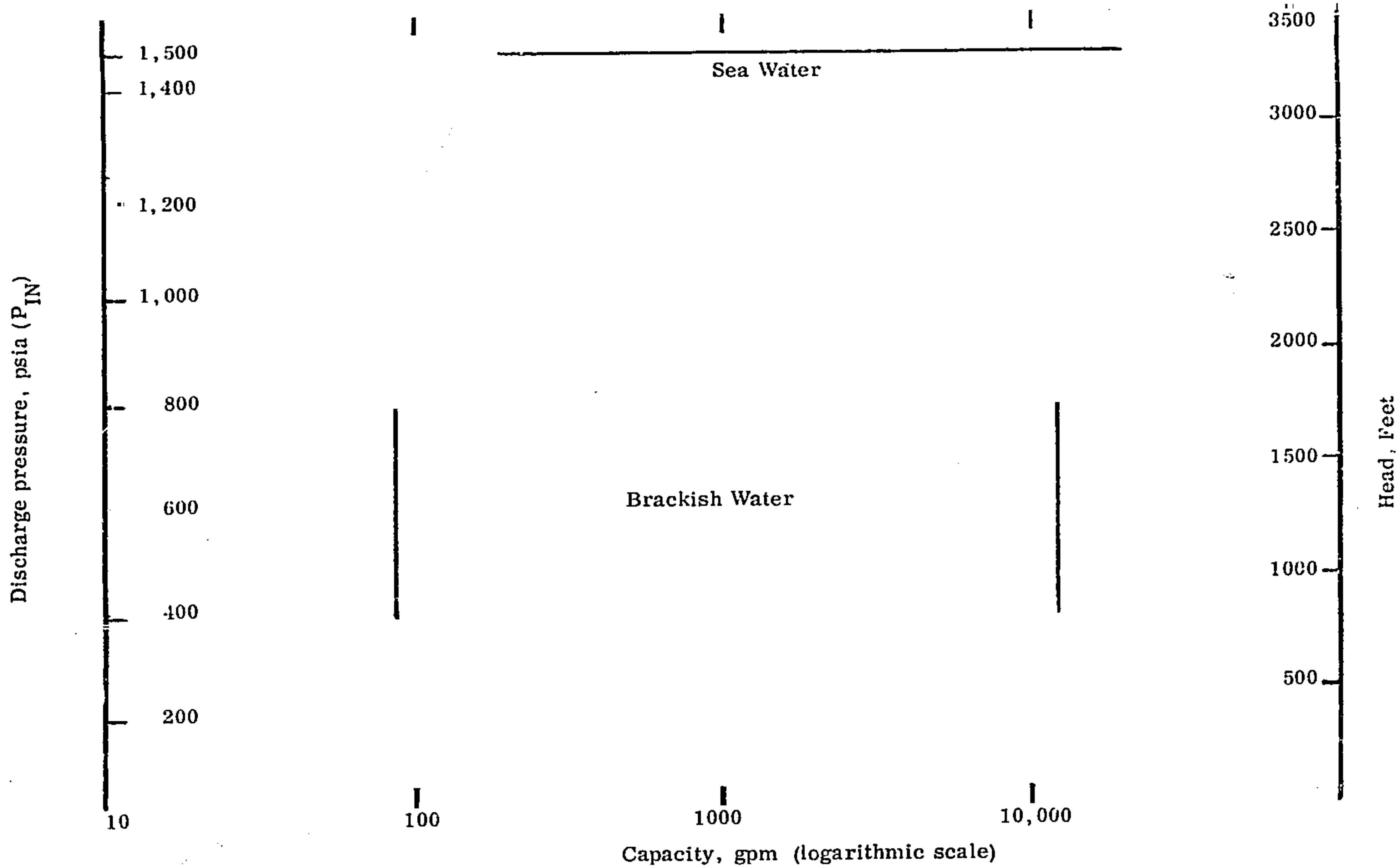


Figure 2.1.1: General Operating Ranges for Pumping Equipment

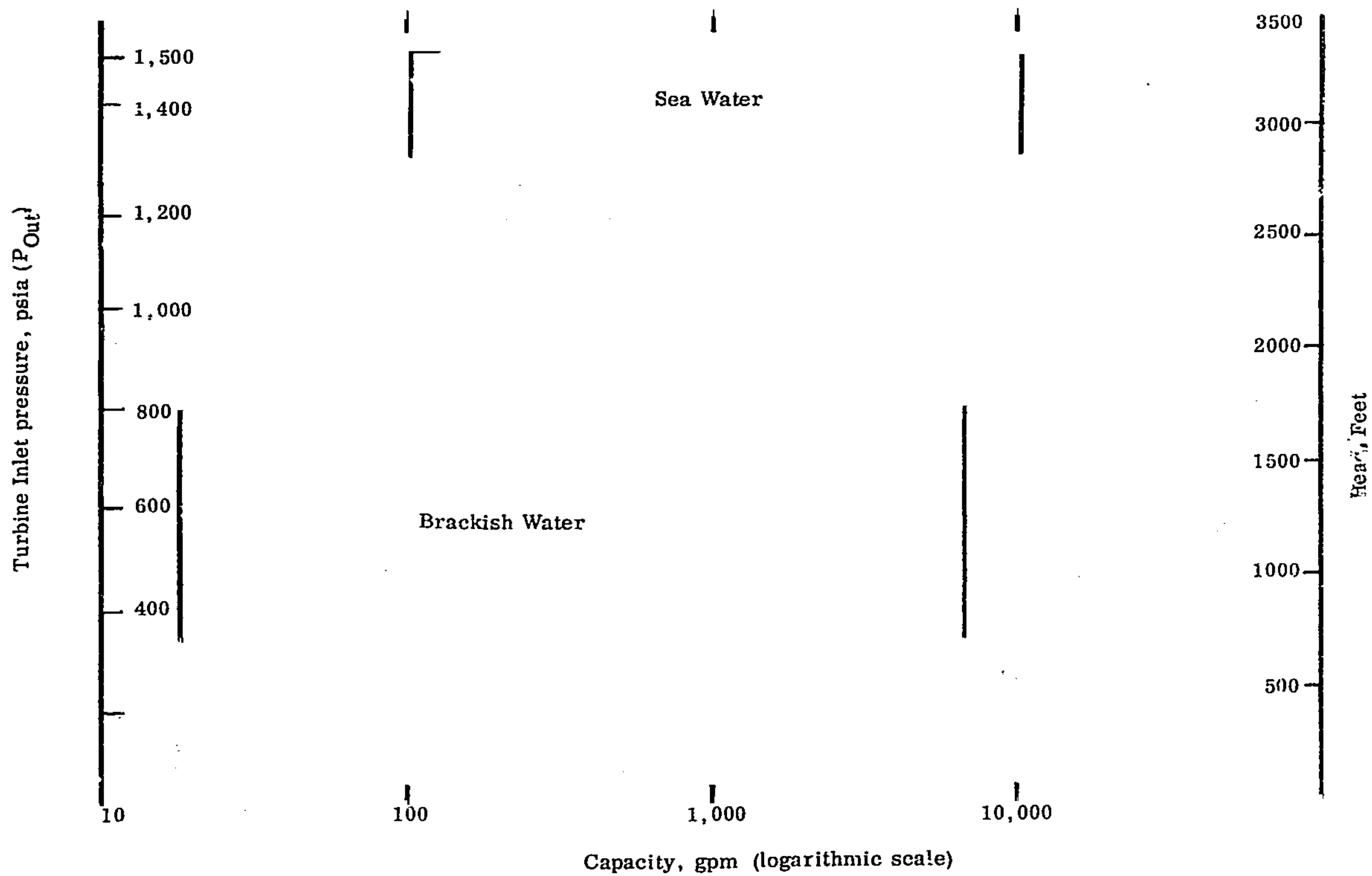


Figure 2.1.2: General Operating Ranges for Energy Recovery Equipment

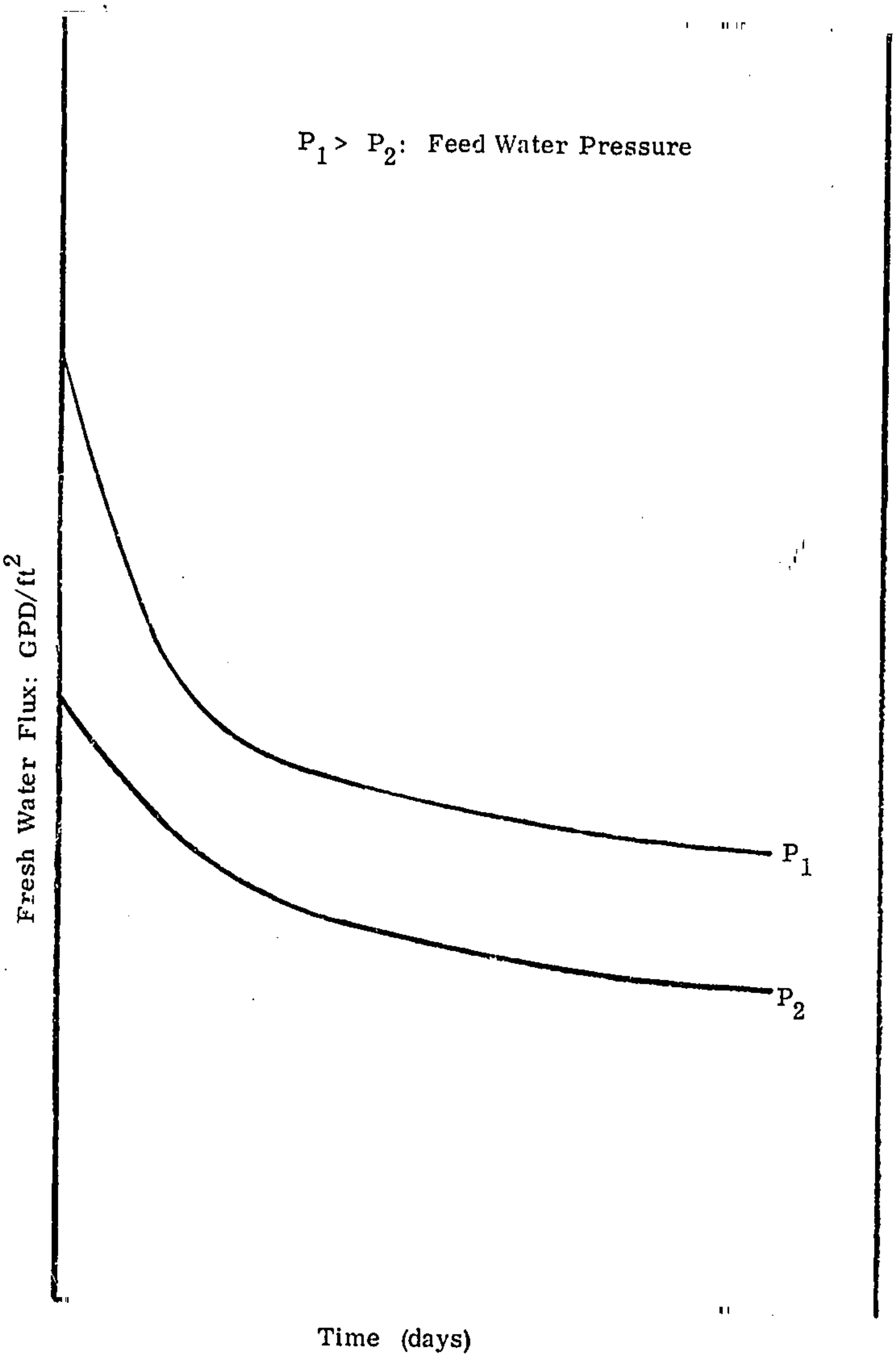
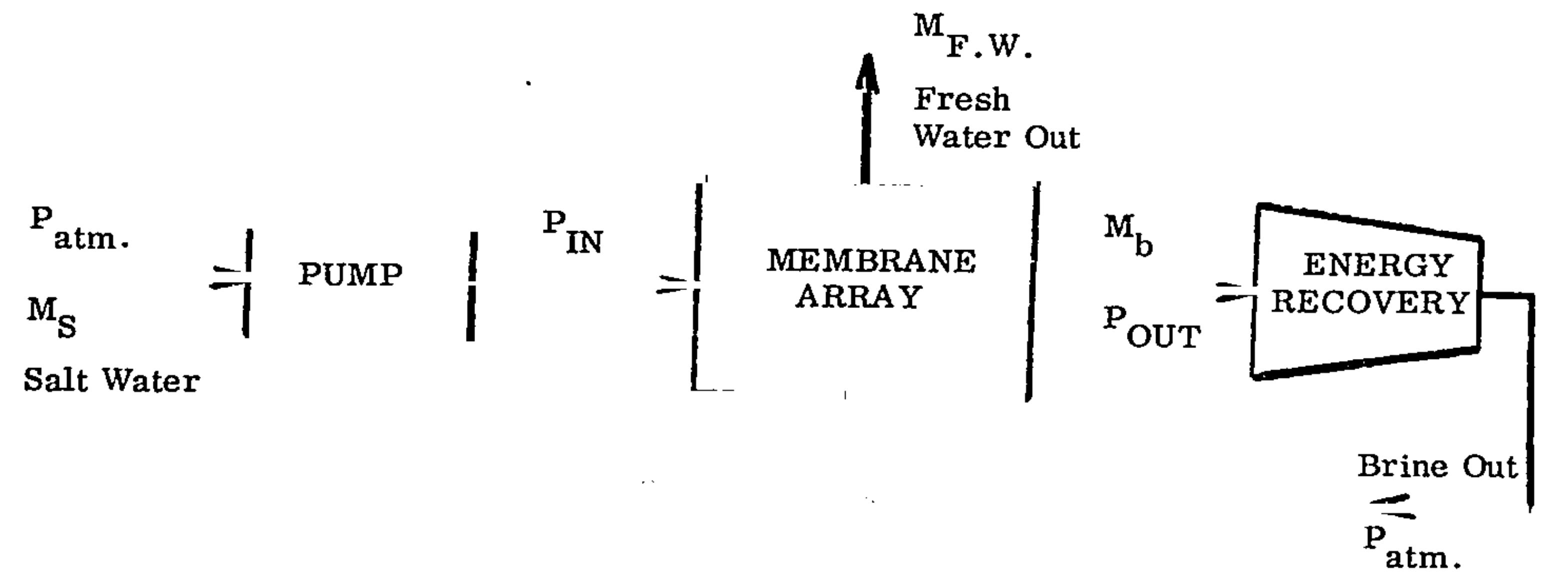


Figure 2. 1. 3: Variations of Fresh Water Flux During a Continuous Run
(from Reference No. 118)



M: Flow Rate or Capacity (GPD)

P: Pressure (psia)

$$M_S = M_{F.W.} + M_b \quad , \quad R = \frac{M_{F.W.}}{M_S}$$

Figure 2.1.4 Schematic Flow Diagram of a Reverse Osmosis Desalination Plant

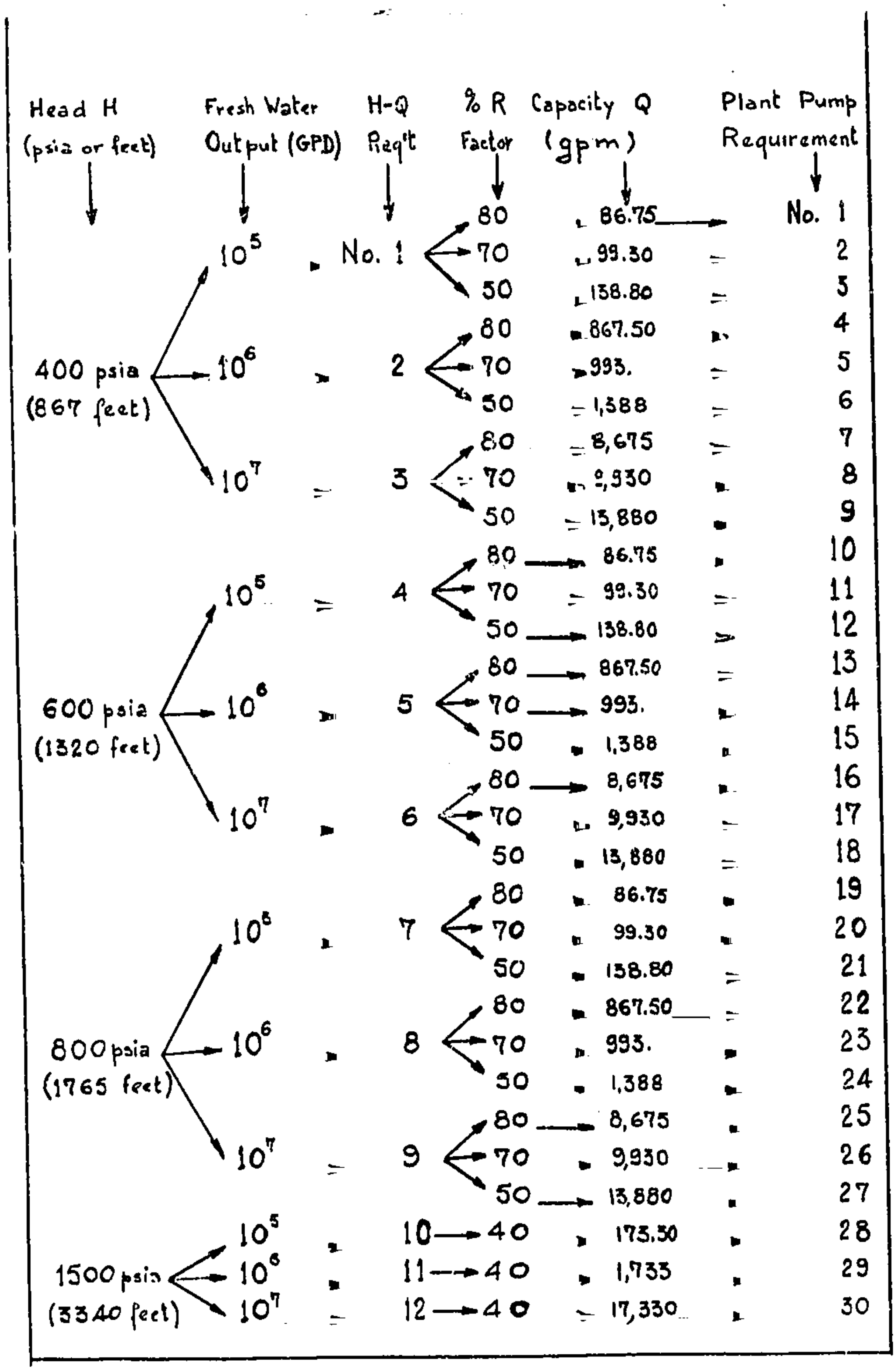


Figure 2.1.5 Standard Plant and Pumping System Network

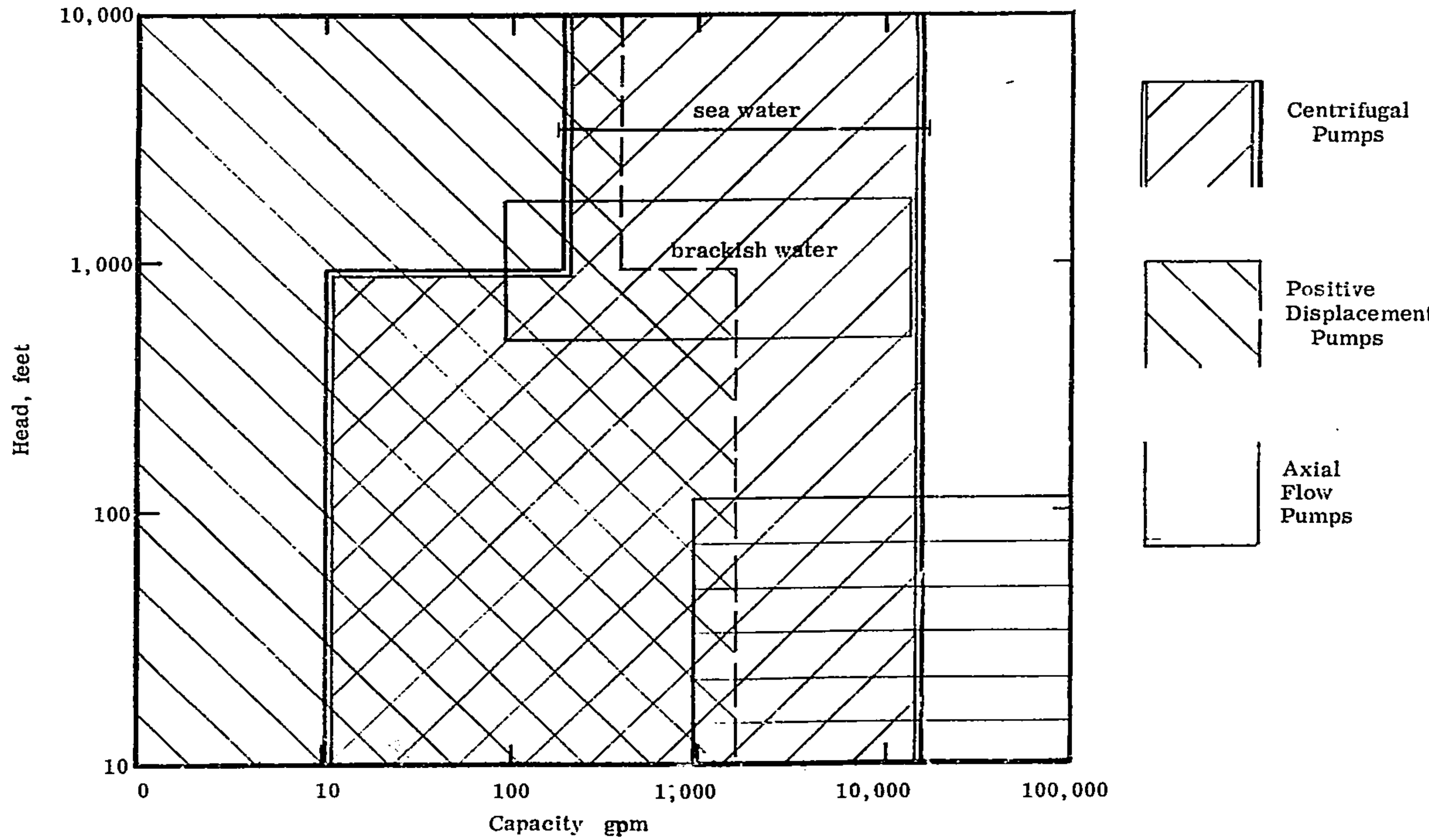
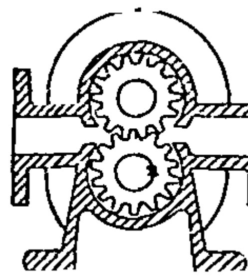
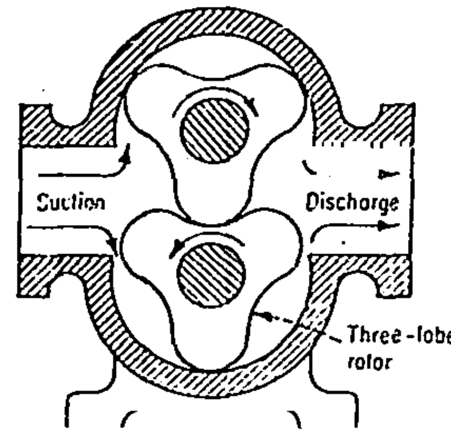


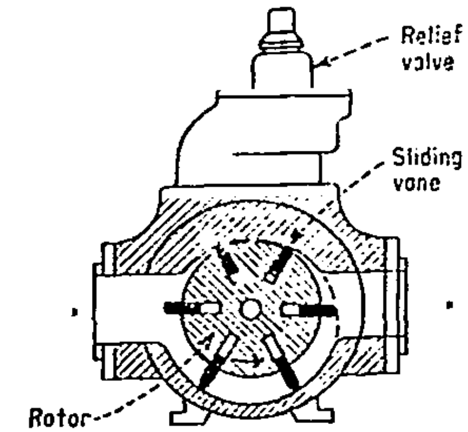
Figure 2.2.1 General Domains of Operation of Standard Pumping Equipment



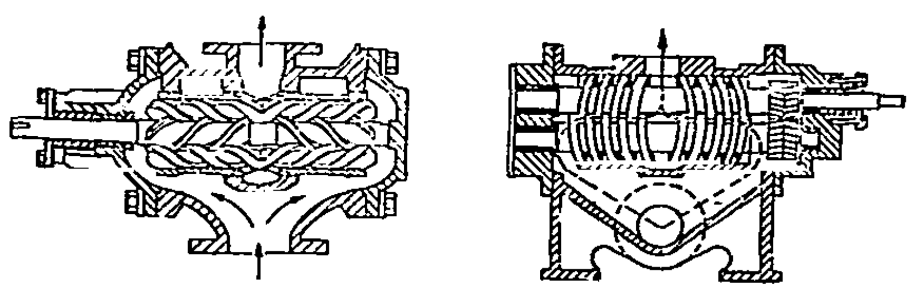
Gear Pump (from Ref. 117)



Lobe Pump



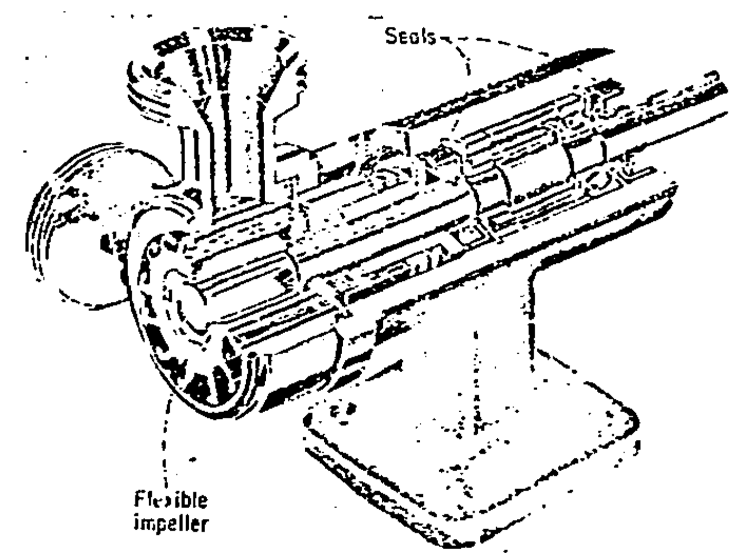
Sliding Vane Pump



Single

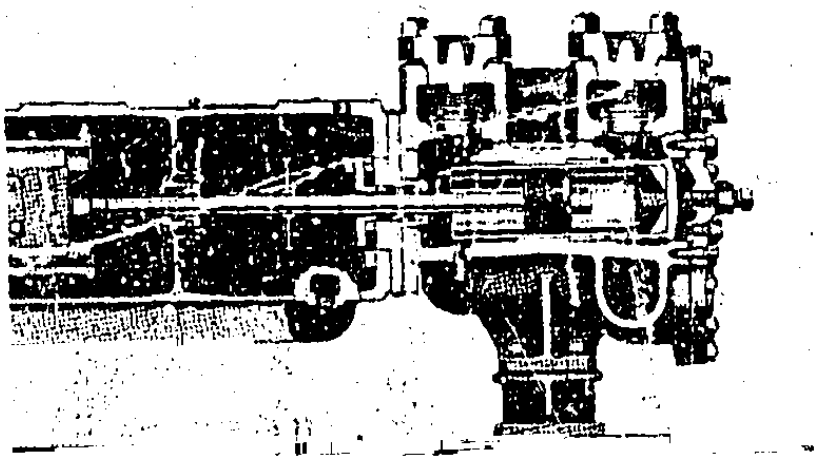
Twin

Screw Pumps (from Ref. 117)

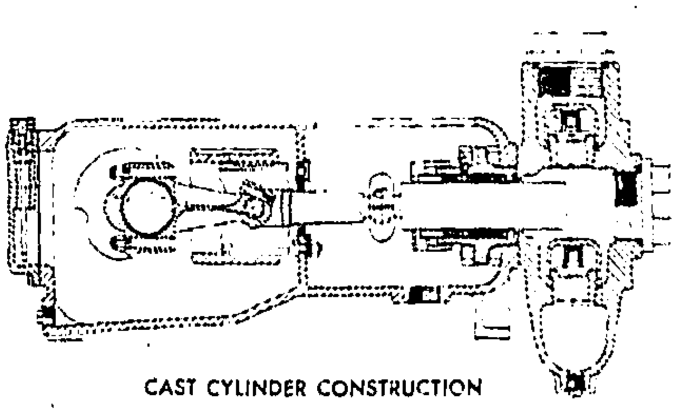


Flexible Impeller

Figure 2.2.2 Various Types of Rotary Pumps

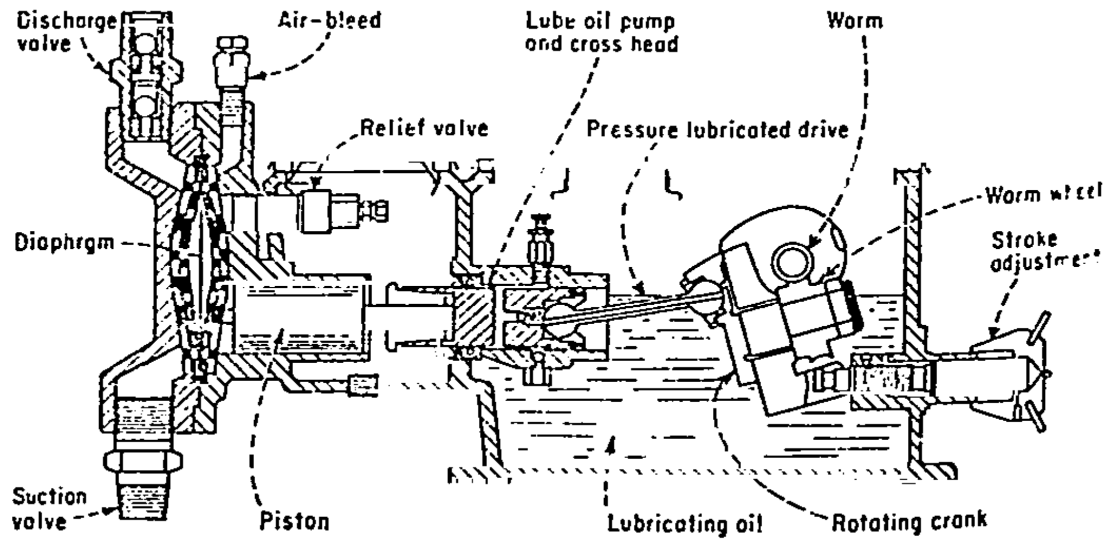


Piston Type

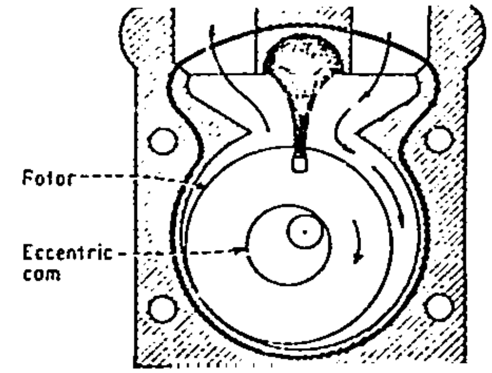


Plunger Type

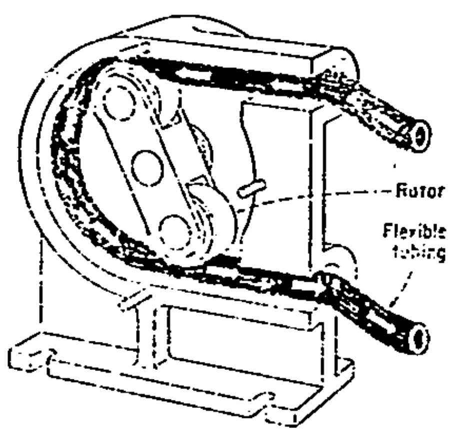
Figure 2.2.3 Reciprocating Pumps
(From G irdner - Denver)



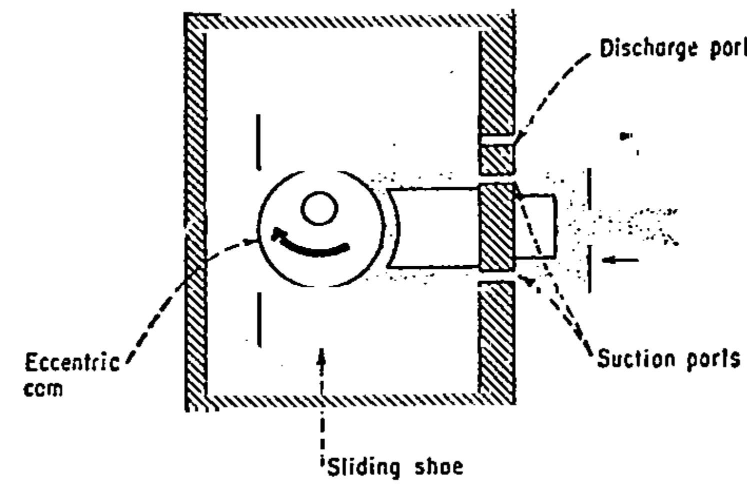
Diaphragm Pump



Eccentric Cam Pump
(constant volume)



Peristaltic Pump



Eccentric Cam Pump
(variable volume)

Figure 2. 2. 4 Other Types of Positive Displacement Pumps

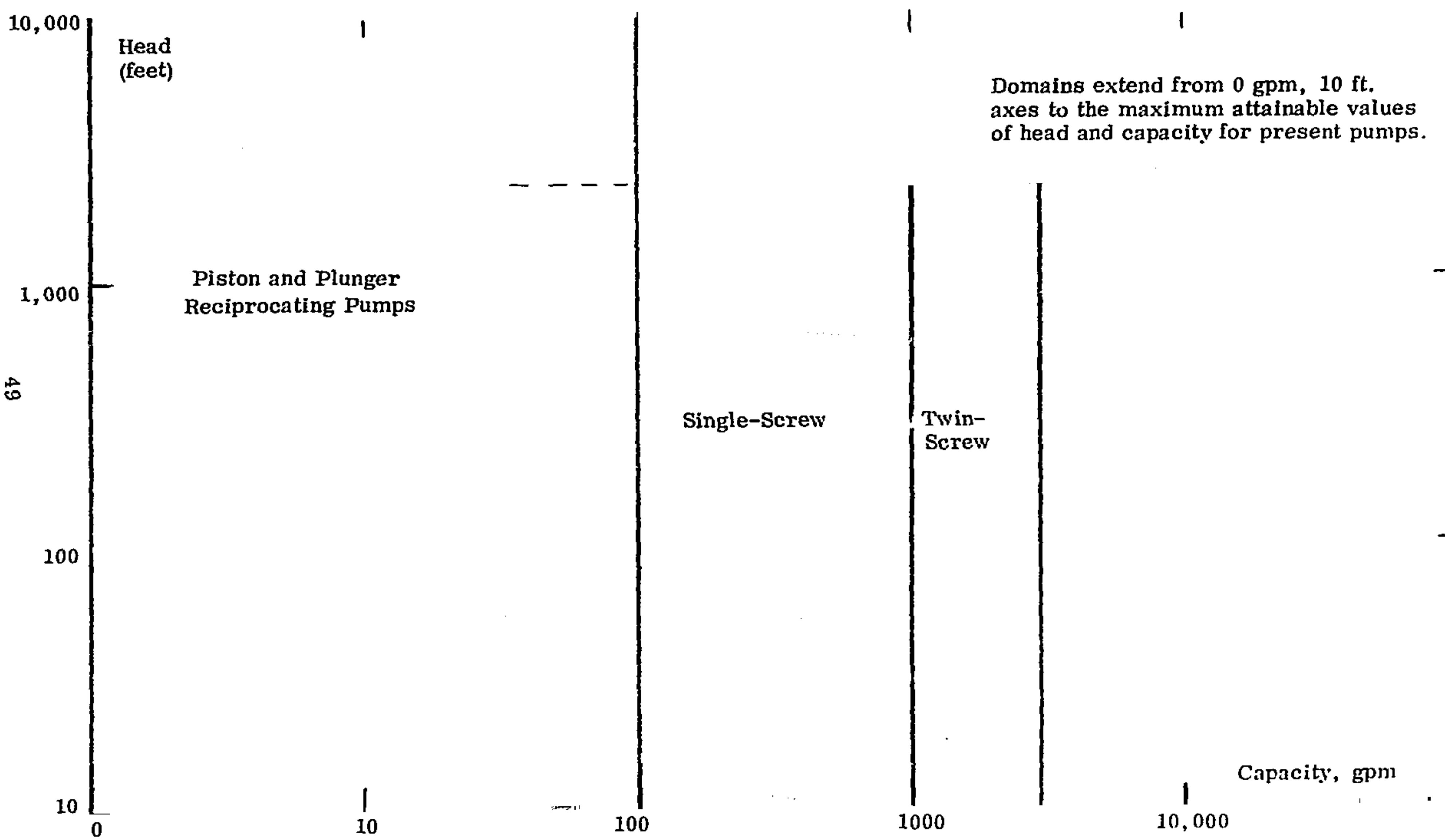


Figure 2.2.5 General Operating Ranges of Positive Displacement Pumps

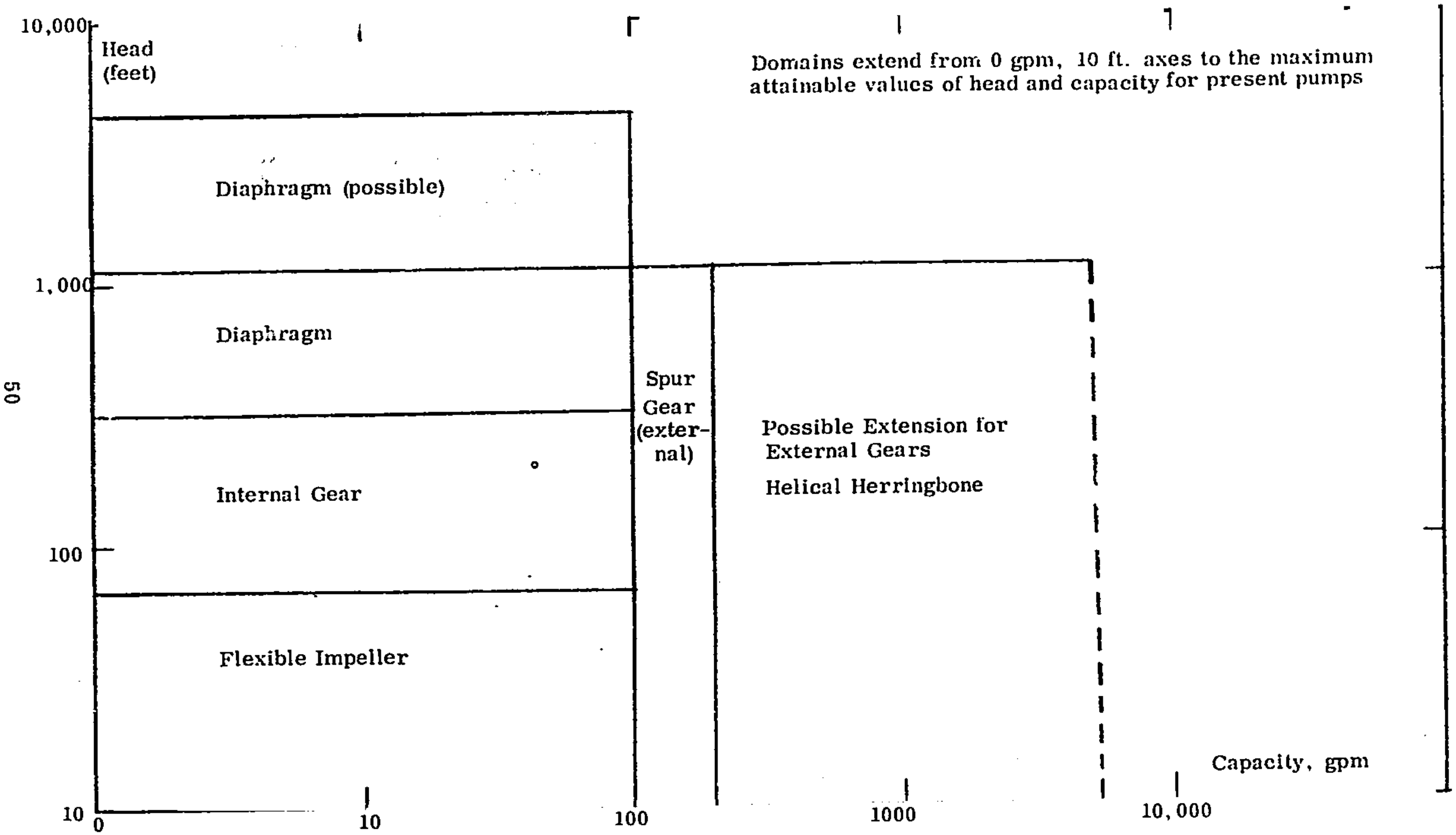


Figure 2.2.6 General Operating Ranges of Positive Displacement Pumps

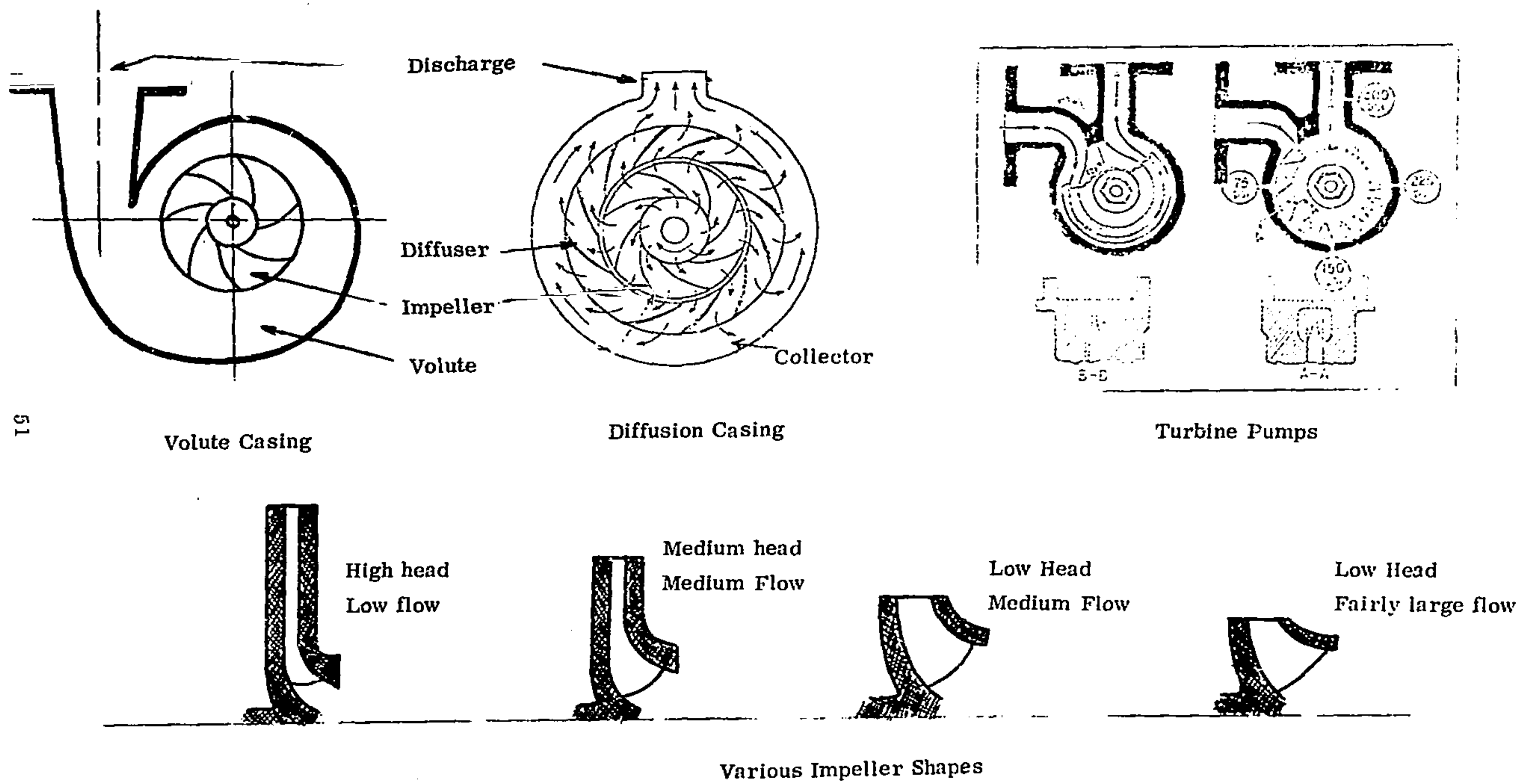


Figure 2. 2. 7 Various Types of Centrifugal Pumps and Impellers

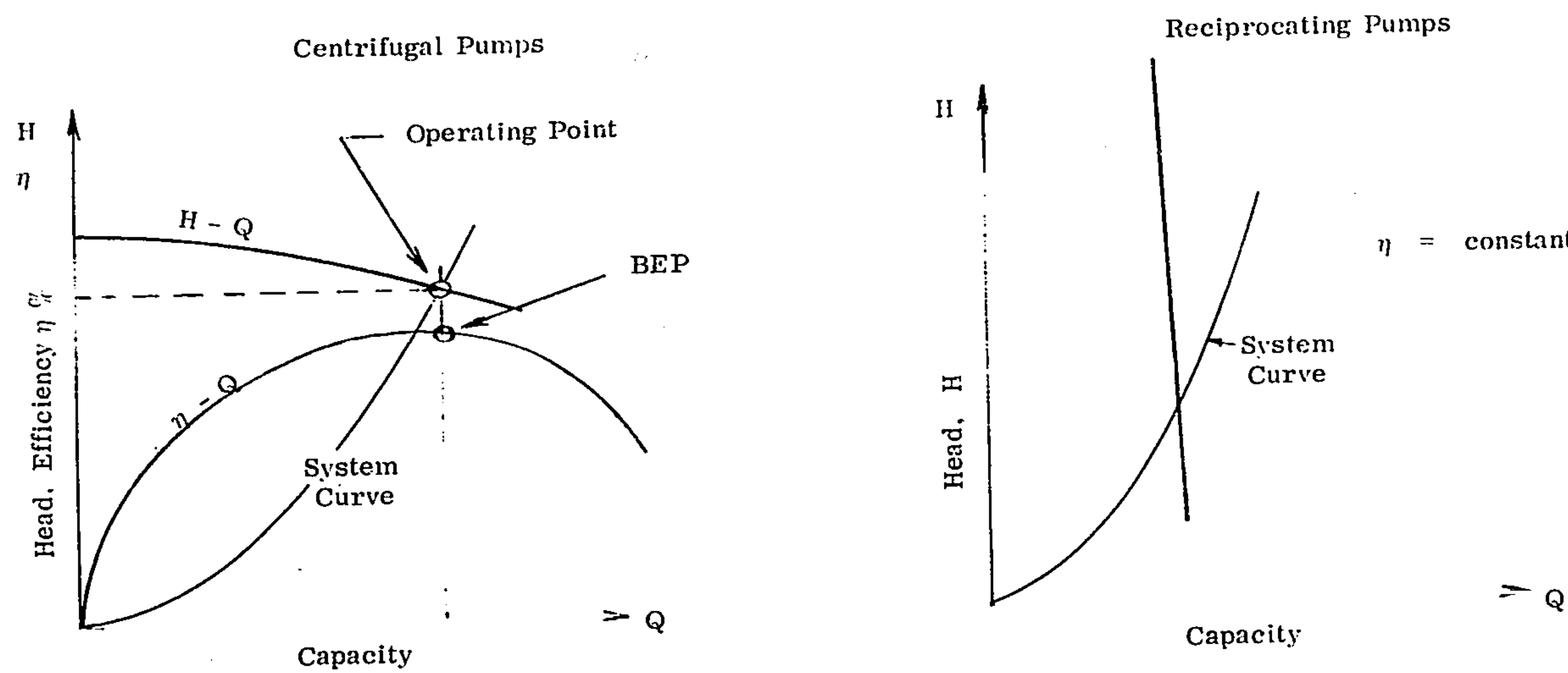
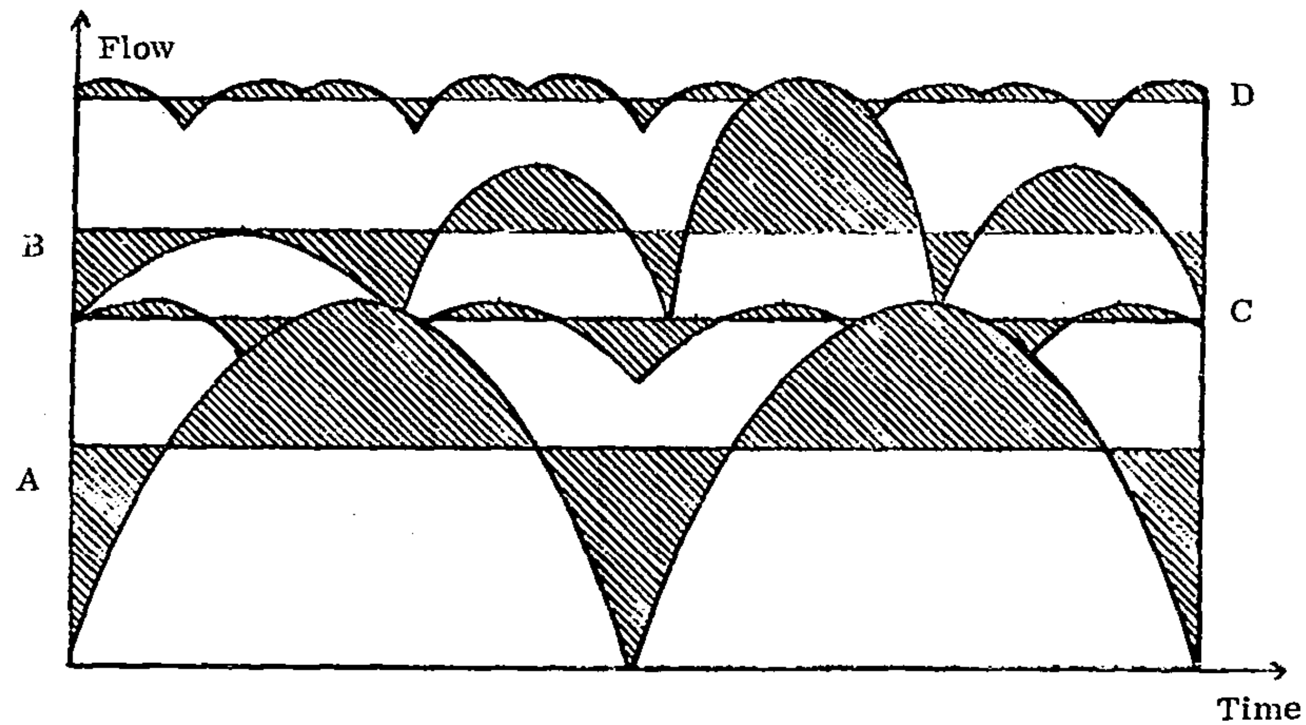


Figure 2. 2. 8 Performance Curve and System Curve



A - Simplex - double acting

Max. rate of flow above mean 59.96%
Min. rate of flow below mean 100.0%

B - Duplex - double acting

Max. rate of flow above mean 26.72%
Min. rate of flow below mean 21.56%

C - Triplex - single acting

Max. rate of flow above mean 6.64%
Min. rate of flow below mean 18.42%

D - Quintuplex - single acting

Max. rate of flow above mean 1.88%
Min. rate of flow below mean 5.63%

Figure 2.2.9 Flow Rate Variations in Positive Displacement Reciprocating Pumps (From Ref 117)

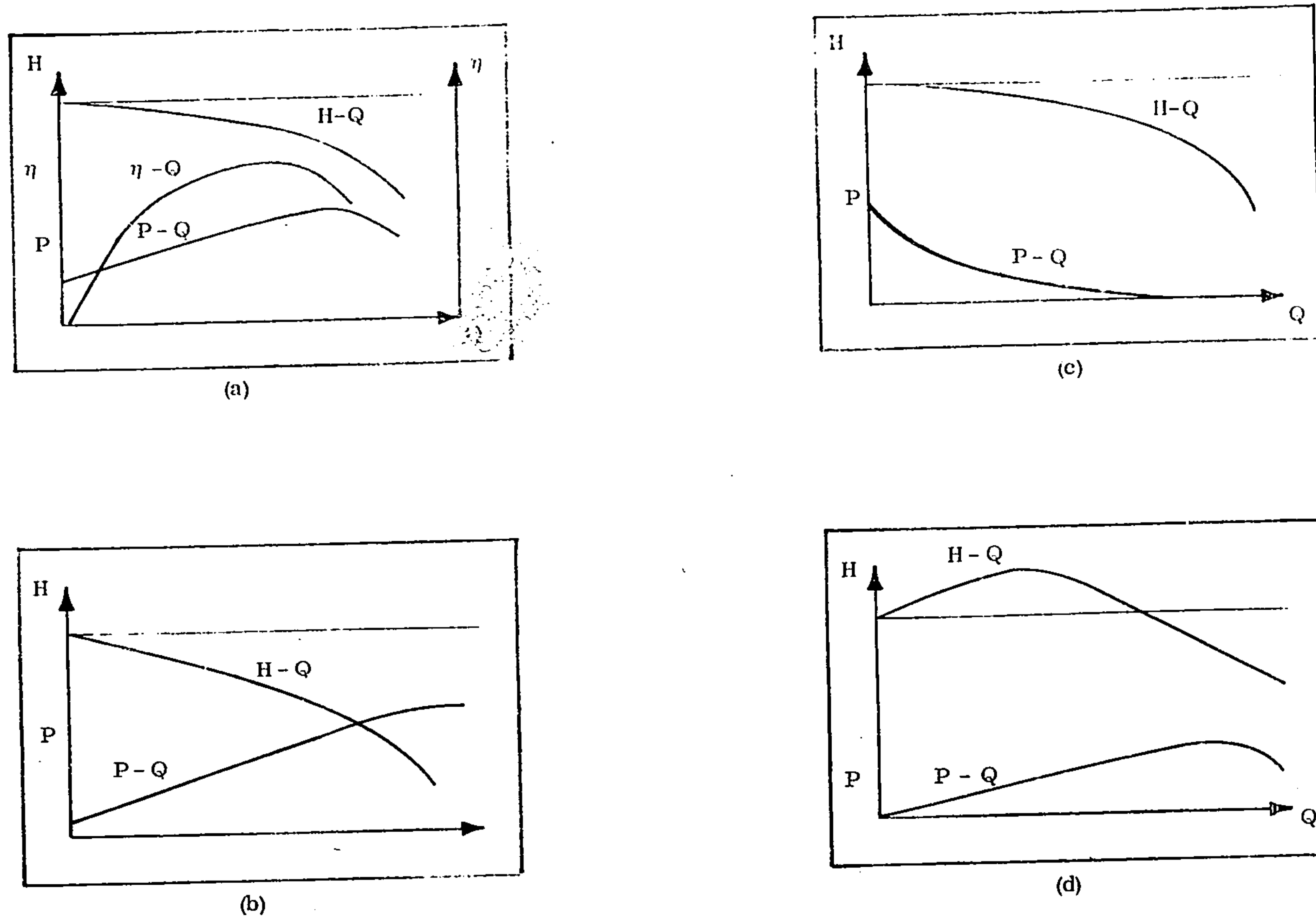


Figure 2. 2. 10 Performance Curves of Centrifugal Pumps

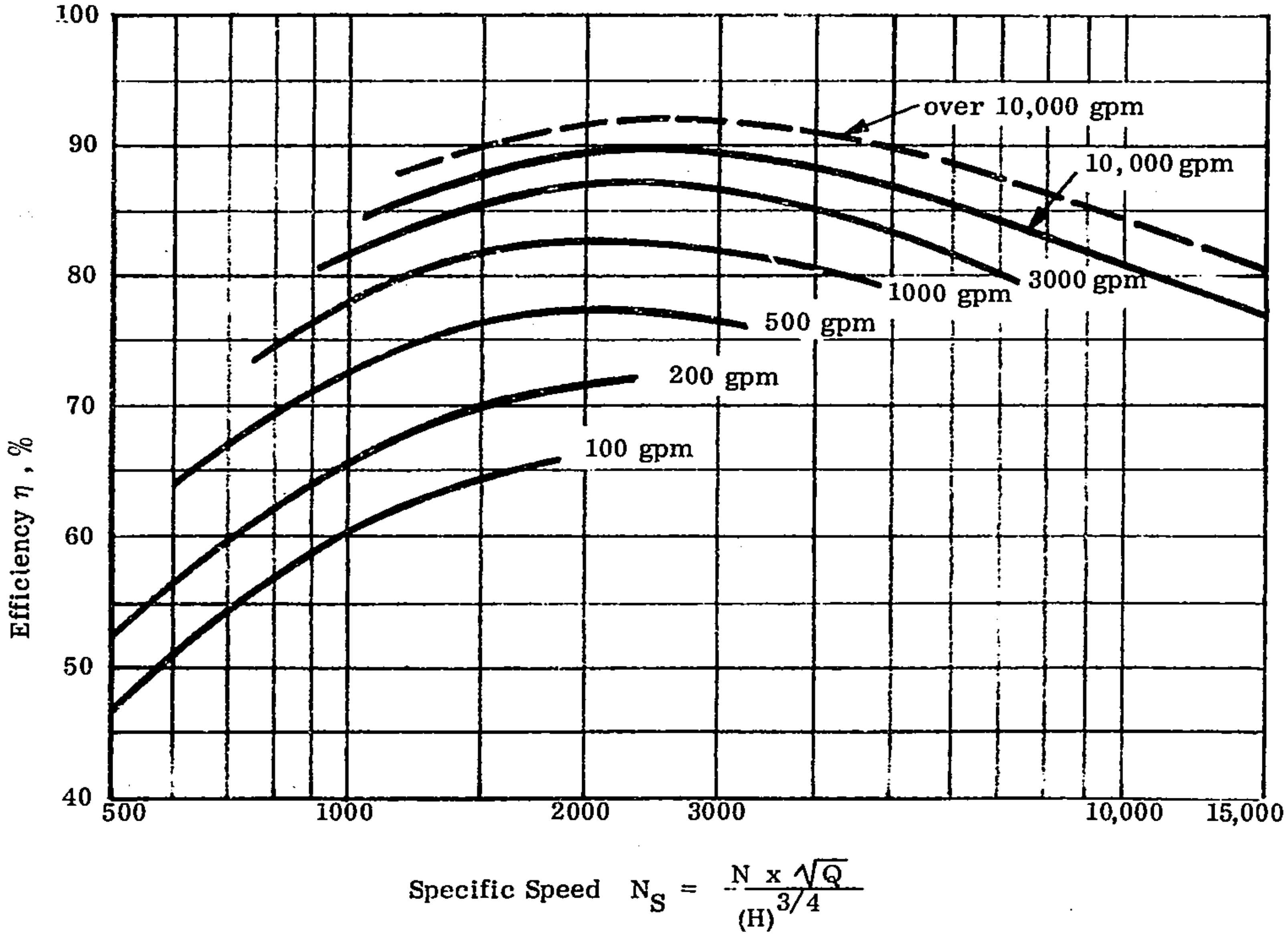
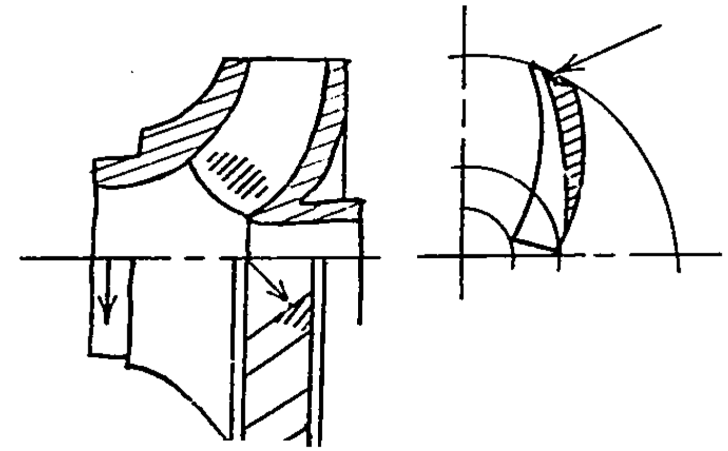
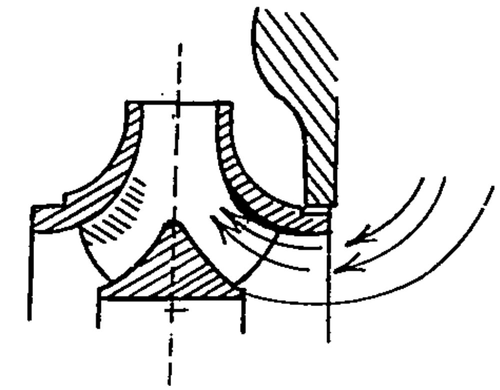


Figure 2. 2. 11 Single Stage Centrifugal Pump Efficiencies (Worthington data) (From Ref. 115)

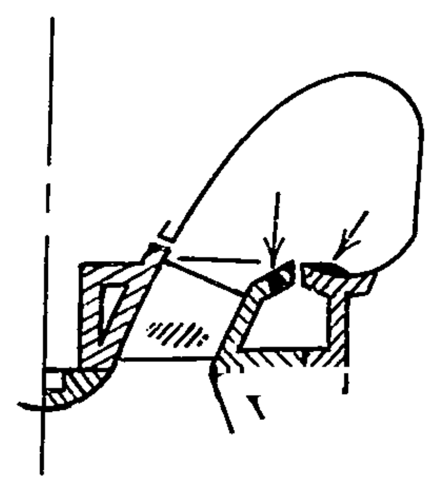


Vane's Blunt Discharge Tip

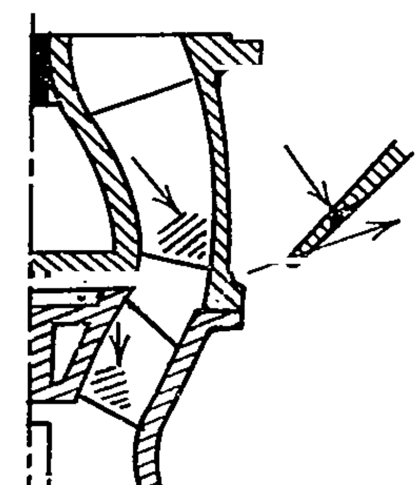


Vane and shroud pitting due to lack of streamlining

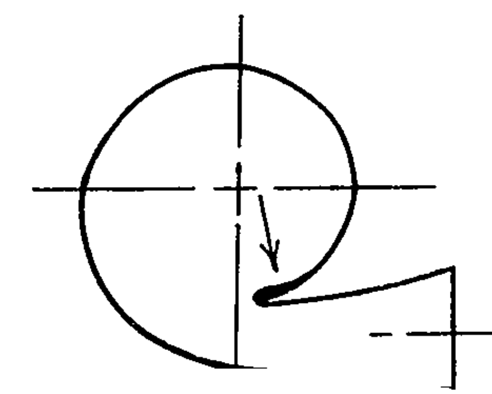
Impeller Cavitation



Lack of streamlining of volute casing

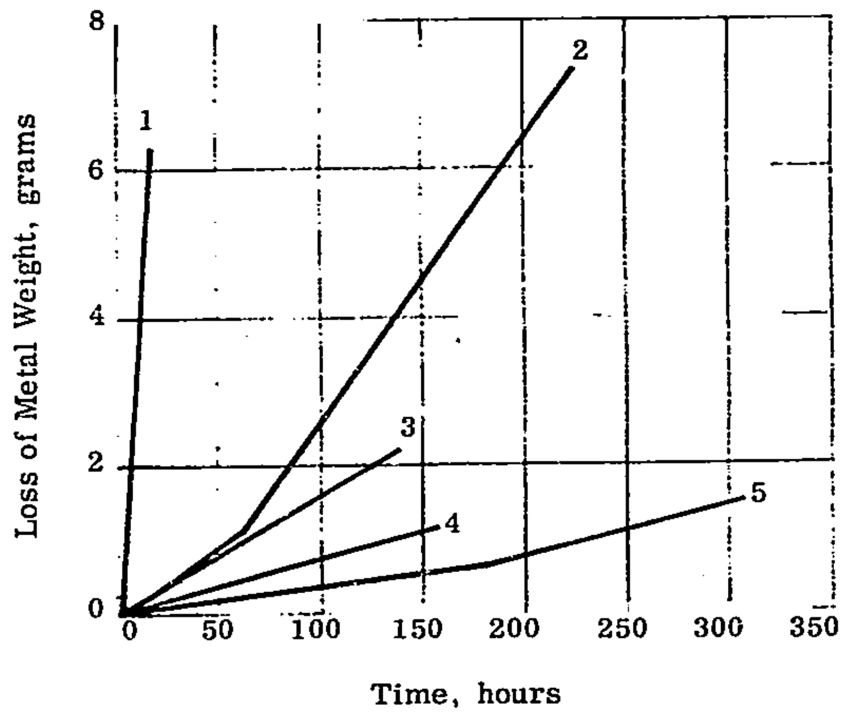


Diffusion Vane, due to bad matching of diffusion vane angle and flow angle



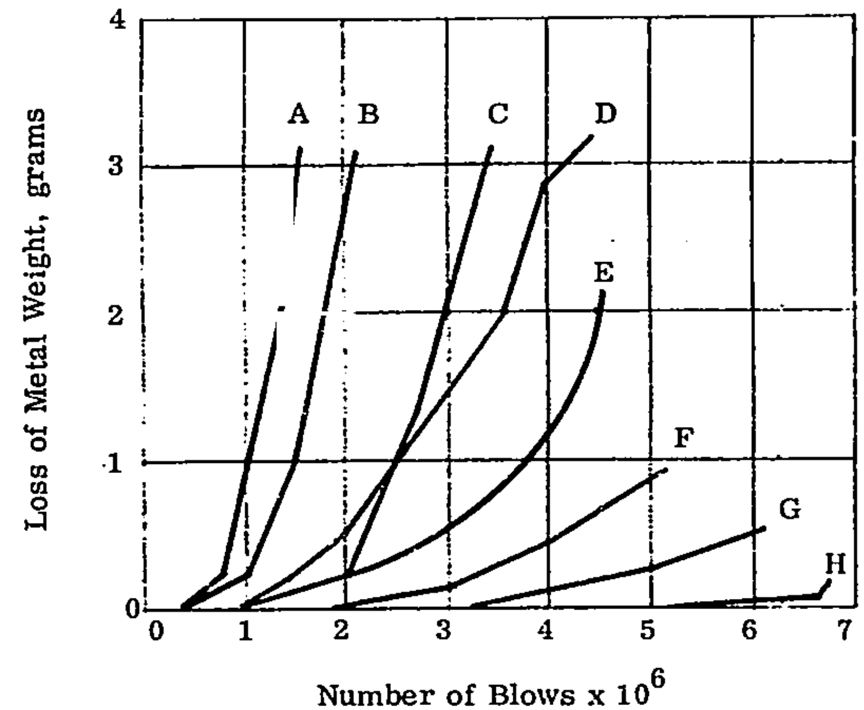
Tongue pitting due to operation @ capacities higher than usual

Figure 2.2.12 Various Causes and Locations of Cavitation Pitting in Centrifugal Pumps (From Ref. 115)



1 Lead
2 Cast Iron
3 Bronze (low speed)
4 Aluminum
5 Steel

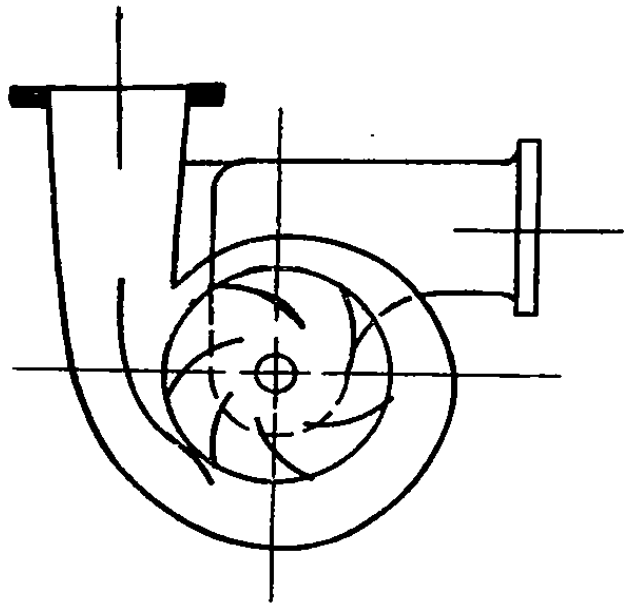
Loss of Metal Due to Cavitation



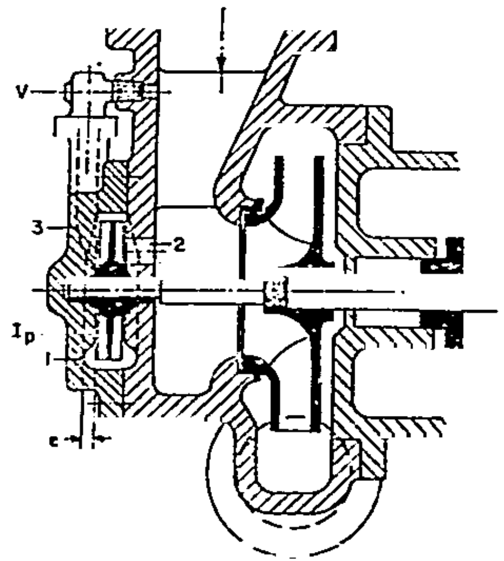
A Nickel Bronze
B Cast Steel
C Aluminum Bronze
D Open-hearth Steel
E Stainless Steel Cast
F Stainless Steel
G Chrome-Vanadium Steel
H Steel H-123 Krupp

Loss of Metal by Jet Drop-Impact

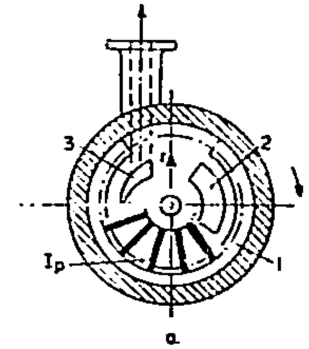
Figure 2.2.13 Material Tested for Cavitation Pitting
(From Ref. 115)



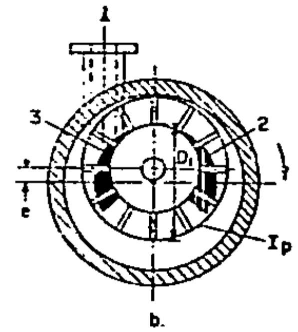
Self-Priming Pump with
Air Separator
(suction chamber)



Self-Priming Pump with
Separator Air Pump

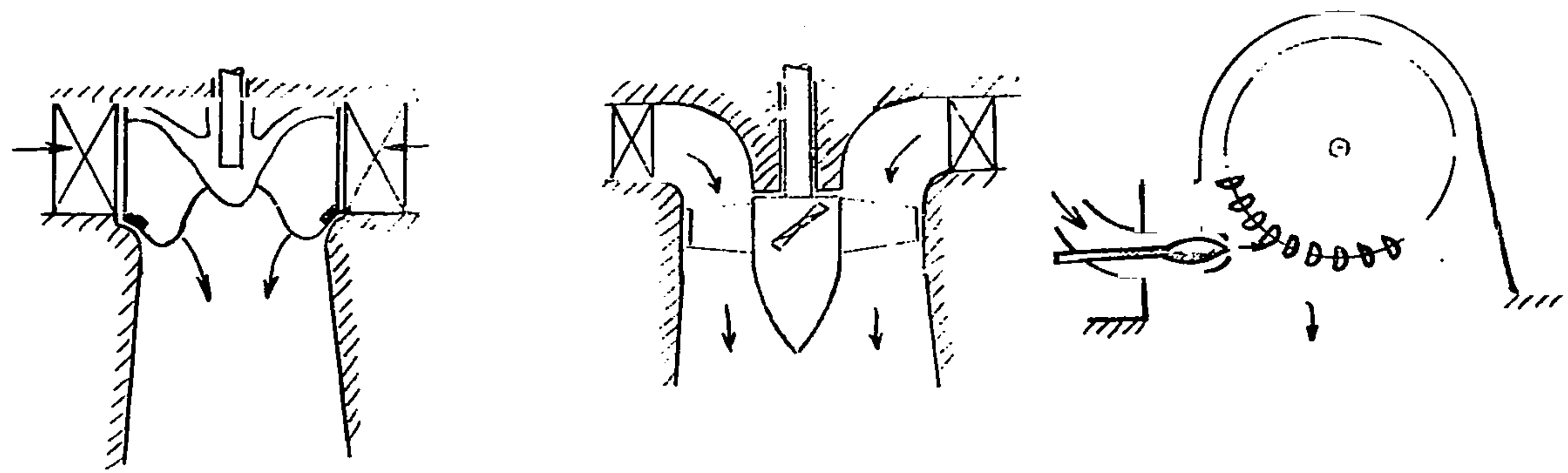


Side-Channel Type



Water-Ring Type

Figure 2.2.14 Self-Priming Systems Used for Centrifugal Pumps
(From Ref 116)



Francis Turbine

Kaplan Propeller Turbine

Pelton Impulse Turbine

Figure 2.4.1 Schematic Cross-Sections of Hydraulic Turbines

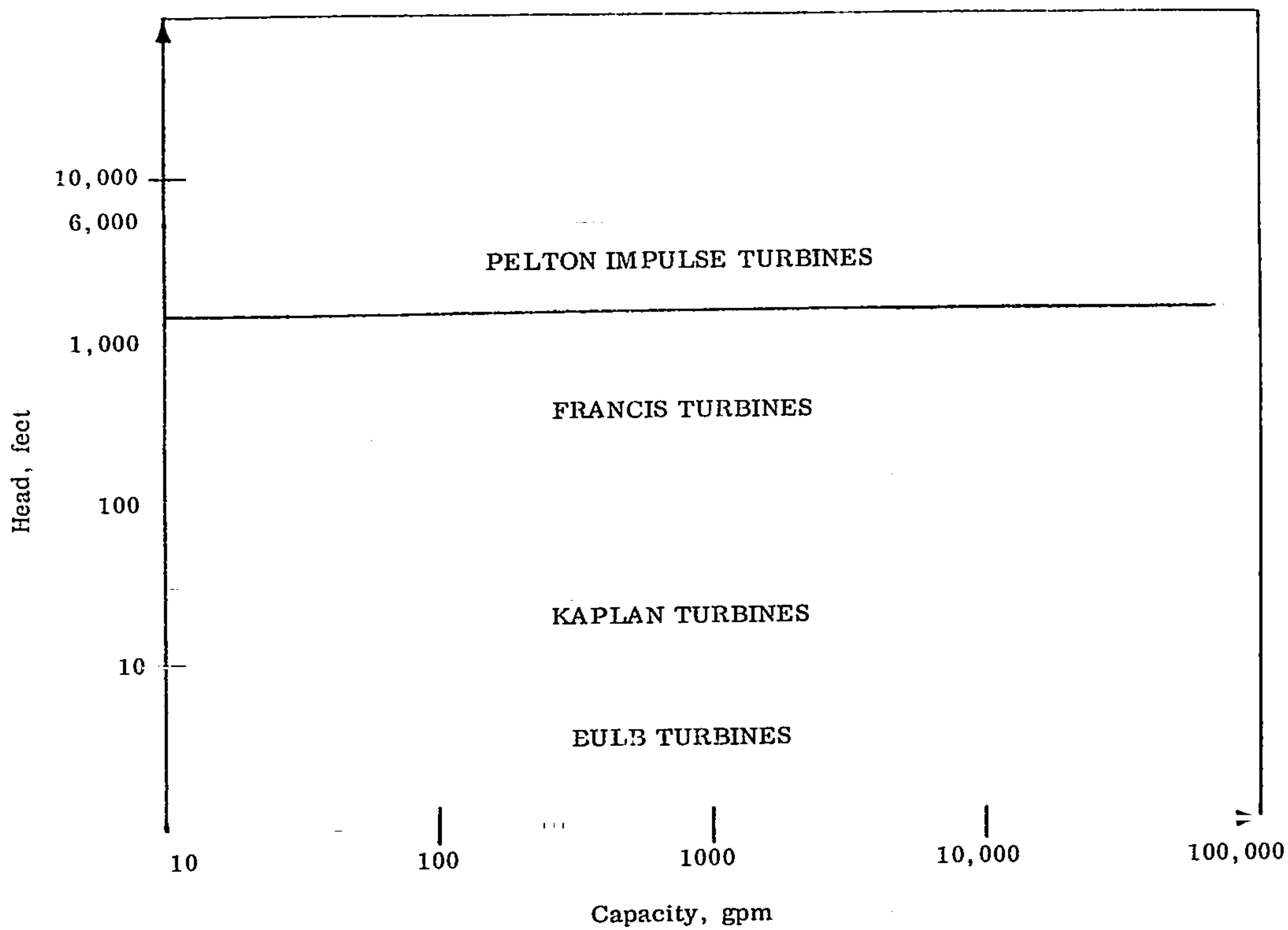


Figure 2.4.2 General Domains of Turbine Operation

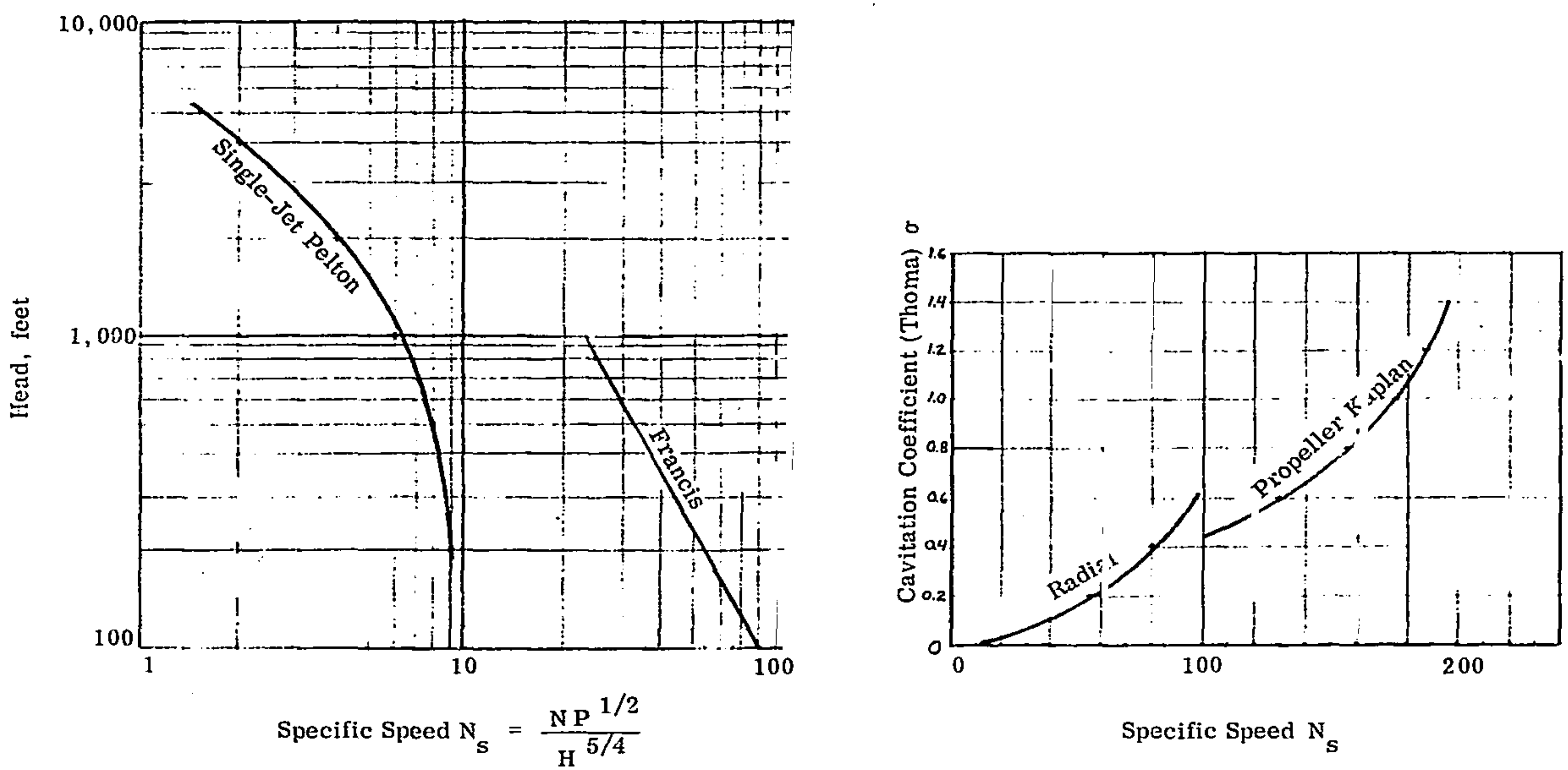
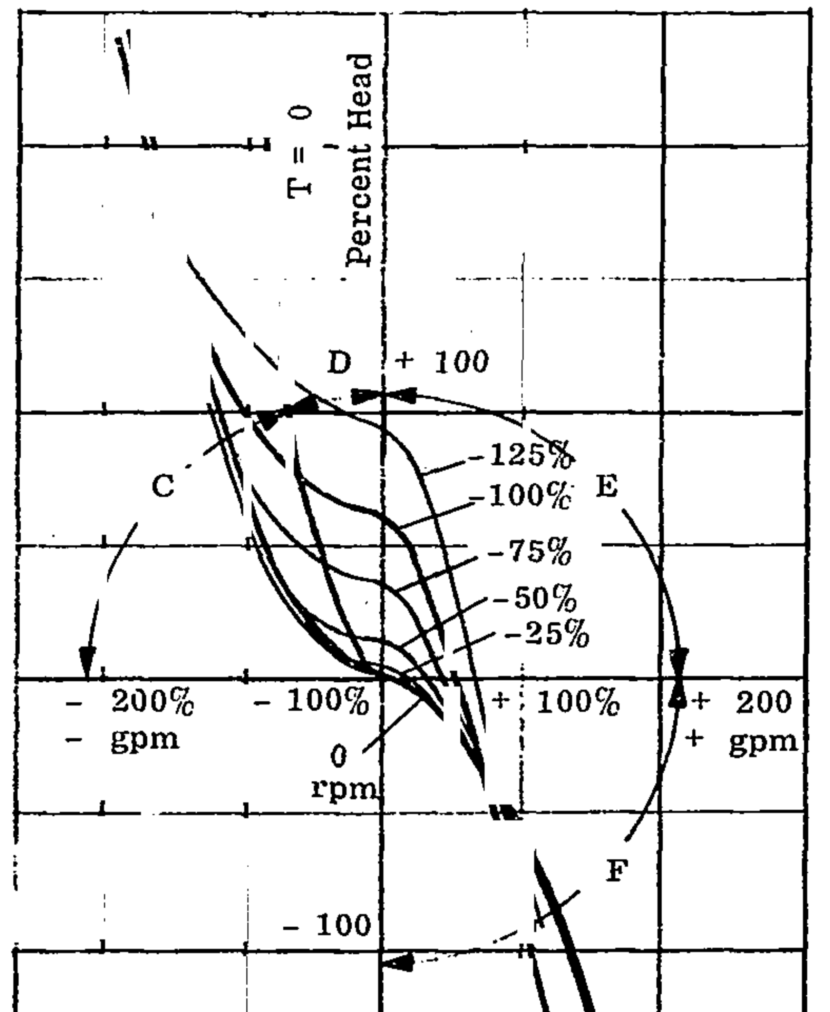


Figure 2.4.3 Head and Cavitation Coefficient Limitation Curves With Specific Speed (From Ref 114)



Region	Function
C	Normal Turbine
D	Energy Dissipation
E	Reverse rpm Pump
F	Energy Dissipation

negative rpm values denote turbine operation

Figure 2.4.4 Pump Operated as a Turbine - Karmann-Knapp Diagram
Double-Suction Pump - N_S = 1800 (From Ref. 115)

Section 3
SELECTION OF AVAILABLE PUMPING EQUIPMENT

3.1 Introduction and Approach

The preceding Section 2 included a general discussion of the types of pumps which are suitable for reverse osmosis desalination plants. A preliminary evaluation of the various types of pumps restricted the field of investigation to positive displacement reciprocating pumps and to multistage centrifugal pumps.

Table 3.1.1 presents a listing of a number of manufacturers of these two types of pumps. These manufacturers were initially requested to supply general catalog information on their standard pumps for high pressure service. A study of these sales catalogs permitted us to develop a detailed questionnaire (cf. Appendix B) that was sent to the manufacturers listed in Table 3.1.2. Those manufacturers who responded to the questionnaire are identified in this table.

The manufacturers' replies indicated that reciprocating pumps are produced in sizes suitable for use in smaller desalination plants (e.g., 10^5 GPD) where their efficiency is superior to that of centrifugal pumps. Vertical centrifugal pumps also are furnished to meet the head and flow requirements of the smaller plants, but their low efficiency makes them less attractive than reciprocating pumps.

Centrifugal pumps are available to cover the middle size plant requirements (10^6 GPD). A number of specific pumps have been selected by the manufacturers to match our various "standard plant" specifications at the pump best efficiency points (BEP). This matching is necessary to avoid maintenance problems created by prolonged operation at off-design conditions.

It was difficult to select available pumps for the larger size plants (10^7 GPD) because their combinations of flow and head requirements are not often encountered in other pump applications. The head needed is too low for standard two-stage centrifugal boiler-feed pumps that will handle the flow. Standard designs of boiler-feed pumps do not permit ready adaption to meet the desalination plant design requirements. Most

MANUFACTURER	TYPE OF PUMP
Allis-Chalmers Mfg. Co. Milwaukee, Wisconsin	Centrifugal Screw
Aurora Pump Division, The N. Y. Air Brake Co. Aurora, Illinois	Centrifugal
Buffalo Forge Co., Pump Division Buffalo, New York	Centrifugal
Byron-Jackson Pumps Inc. Los Angeles, California	Centrifugal Reciprocating
DeLaval Turbine, Inc. Trenton, New Jersey	Centrifugal Reciprocating
Escher-Wyss Germany (New York, N.Y. Office)	Centrifugal
Fairbanks-Morse, Pump Division Colt Industries Kansas City, Kansas	Centrifugal Screw
Gardner-Denver Co. Quincy, Illinois	Centrifugal Reciprocating
Ingersoll-Rand Company Phillipsburg, New Jersey	Centrifugal Reciprocating
Japan Steel Works Ltd. Tokyo, Japan	Centrifugal Reciprocating
Kobe, Inc. Huntington Park, California	Reciprocating
Lawrence Pumps Lawrence, Mass.	Centrifugal Screw
Manton-Gaulin Mfg. Co. Inc. Everett, Mass.	Reciprocating
F.E. Myers & Bros. Co. Ashland, Ohio	Centrifugal Screw
New York Air Brake Co. New York, N.Y.	Reciprocating Rotary

Table 3.1.1 Pump Manufacturers

MANUFACTURER	TYPE OF PUMP
Pacific Pumps Inc. Huntington Park, California	Centrifugal
Peerless Pump Division, FMC Corp. Los Angeles, California	Centrifugal Screw
Rateau Ets France	Centrifugal Reciprocating
Roy E. Roth Co., Turbine Pump Division Rock Island, Illinois	Centrifugal
Sulzer Bros., Inc. Germany (New York, N.Y. Office)	Centrifugal Screw
Tuthill Pump Co. Chicago, Illinois	Rotary
Vickers Inc., Division of Sperry-Rand Corp. Troy, Michigan	Positive Displacement
Warren Pump Inc. Warren, Mass.	Centrifugal Reciprocating, Screw
Worthington Corp. Harrison, New Jersey	Centrifugal Reciprocating

Table 3.1.1 Pump Manufacturers (Continued)

Pump Manufacturers	Answer
Mr. J. Dakin Process Pump Sales Worthington Corporation 401 Worthington Avenue Harrison, New Jersey	Information provided.
Mr. D. F. Crego Product Manager, Factory Sales Group Custom Pump & Compressor Dept. Allis-Chalmers Company 884 South 70 Street Milwaukee, Wisconsin 53201	Some information provided.
Mr. Robert Fornesi Cameron Pump Division Ingersoll-Rand Company Phillipsburg, New Jersey	Information provided.
Marketing Manager DeLaval Turbine, Inc. 820 Nottingham Way Trenton, New Jersey 08602	No information
Mr. G. V. Arata Industrial Equipment Division Baldwin-Lima-Hamilton Corporation Philadelphia, Pennsylvania 19142	No information.
Mr. Paul H. Jamison Manager of Industrial Sales Kobe, Incorporated 3038 East Slauson Avenue Huntington Park, California 90256	Information provided.
Mr. Vollendorf Vice President Marketing Fairbanks-Morse Pump Division, Colt Industries 3601 Kansas Avenue Kansas City, Kansas	Partial information provided.
Mr. Wagner Sulzer Brothers Limited Department 4 8401 Winterthur Switzerland	No information.

Table 3.1.2 Manufacturers Receiving the Questionnaire on Pumping Equipment

Pump Manufacturers	Answer
Mr. R. Young Atlantic District Manager Peerless Pump, FMC Corporation 76 Beaver Street New York, New York 10005	Information provided
Mr. Robert Ganz Marketing Manager Gardner-Denver Company 100 Williamson Street Quincy, Illinois 62301	Information provided.
Mr. G. Murphy Byron-Jackson Pumps, Inc. 10 Kearney Road Needham, Massachusetts 02194	Information provided
Mr. James Hope Pacific Pumps Huntington Park California	No information.

Table 3.1.2 Manufacturers Receiving the Questionnaire on Pumping Equipment (continued)

manufacturers recommended the use of smaller-flow pumps operating in parallel or preparation of special designs for the 10^7 GPD plant size. These recommendations are discussed in Section 10.

3.2 Manufacturers Contacted

The questionnaire of Appendix B was sent to the manufacturers listed in Table 3.1.2. Several of the manufacturers were also visited in order to obtain more information. The manufacturers recommended particular pumps from their product lines which they considered to be best suited for the different plants described in Figure 2.1.5. Their selection was based on operation of the equipment at BEP and they chose the most efficient pump available for each application.

Table 3.2.1 indicates, for each plant, the different pumps which have been offered by the manufacturers together with their most important performance characteristics. A final figure (Figure 3.2.1) summarizes the ranges of speeds and efficiencies for pumps that meet the various plant design requirements of head and capacity.

3.3 Operation of Selected Equipment

Instructions for the operation of hydraulic machinery are always provided by the manufacturer when a machine is delivered. These instructions are specifically applied to each unit. However, general ground rules can be stated for safe operation of reciprocating pumps and centrifugal pumps.

3.3.1 Reciprocating Pumps

Before starting a new pump, its crankcase should be cleaned and filled with oil to the proper level. When possible, the pump should be run at reduced speed but not run below the minimum speed recommended. The pump should be operated for some hours at a low discharge pressure. The oil level and oil pressure must be checked frequently. The pump may then be brought up to full speed and full discharge pressure gradually. The pump must not be operated at speeds exceeding rated speed or below minimum speed. The specified horsepower and delivery pressure limits must not be exceeded.

To assure proper lubrication, the pump must be driven in the direction indicated on the frame.

Problems to watch for during operation include the following:

- Undue heating or abnormal noise

- Air leaks in the suction line or clogging of inlet filters, if any

- Abnormal vibration caused by improper suction conditions

- Valve leaks contributing to a highly fluctuating discharge pressure

A reciprocating pump must always be protected from excess pressure by a relief safety valve which is to be installed near the pump in the discharge manifold. This valve should be set to operate at about 1 1/4 times the discharge pressure.

3.3.2 Centrifugal Pumps

A number of general precautions must be taken to insure satisfactory operation of a centrifugal pump during the start-up period and operating periods.

Check that all external surfaces are clean and priming conditions are satisfied. Then, test the driver for rotation; the arrow on the pump casing will show the correct rotation. Check the alignment of pump and driver.

The bearings must be lubricated before startup. Guidelines for various types of bearings follow:

- Kingsbury Thrust Bearings: When started for the first time, a generous amount of oil should be poured into the bushing where the thrust shoes are located.

- Grease-Lubricated Ball Bearings: The bearings are usually packed with the correct amount of grease before leaving the factory. Remove the housing cover to check for deterioration and replace according to instructions if necessary.

- Oil-Lubricated Ball Bearings: Flush out the bearings and housings with kerosene or carbon tetrachloride. Fill the reservoir to the proper level and see that the oil is maintained at the correct level while in operation.

Before starting the pump, rotate the unit by hand through at least one complete revolution to make sure that all parts are free. The pump must always be started against a closed gate valve. The starting procedure should be as follows (Ref. 26) : (Figure 3.3.1. illustrates typical external services involved in startup)

1. Prime the pump, open the suction valve, and close the drains to prepare the pump for operation.

2. Open the valve in the cooling-water supply to the bearings.

3. Open the valve in the cooling-water supply if the stuffing boxes are water-cooled.

4. Open the valve in the sealing liquid supply if the pump is so fitted.

5. Open the valve in the recirculating line if the pump should not be operated against dead shut-off.

6. Start the motor.

7. Open the discharge valve slowly.

8. Observe the leakage from the stuffing-boxes and adjust the sealing liquid valve for proper flow to insure the lubrication of the packing, If the packing is new, do not tighten up on the gland immediately, but let the packing run in before reducing the leakage through the stuffing-boxes.

9. Check the general mechanical operation of the pump and motor.

10. Close the valve in the recirculating line once there is sufficient flow through the pump to prevent overheating.

On multistage pumps having a balancing-chamber leakoff, make sure that the leak-off line is not restricted or closed at any time during operation.

Periodic inspections should be made while the pump is running. The stuffing-box should be adjusted so that there is enough leakage to lubricate the packing. The stuffing-box pressure should not exceed the limit imposed for efficient operation. Too much friction due to stuffing-box overpressure may result in a large drop of the pump efficiency.

Stuffing-Boxes and Packings: Severe suction conditions will require special stuffing-box arrangements and special types of packing. Stuffing-box arrangements also vary with the application for which the pump is used. The bearings should be checked for oil level, oil circulation and adequate cooling-water. The by-pass line must be opened when the pump is operated at shut-off or less than 20% of normal capacity.

Generally, the steps followed to stop a pump which can operate against a closed gate valve are as follows:

1. Open the valve in the recirculating line.

2. Close the gate valve.

3. Stop the motor.

4. Close the valve in the cooling-water supply to the bearings and to water-cooled stuffing boxes.

5. If the sealing liquid supply is not required while the pump is idle, close the valve in this supply line.

6. Close the suction valve and open the drain valves as required by the particular installation or if the pump is to be opened up for inspection.

In case the pump is of a type which does not permit operation against a closed gate valve, steps 2 and 3 are reversed.

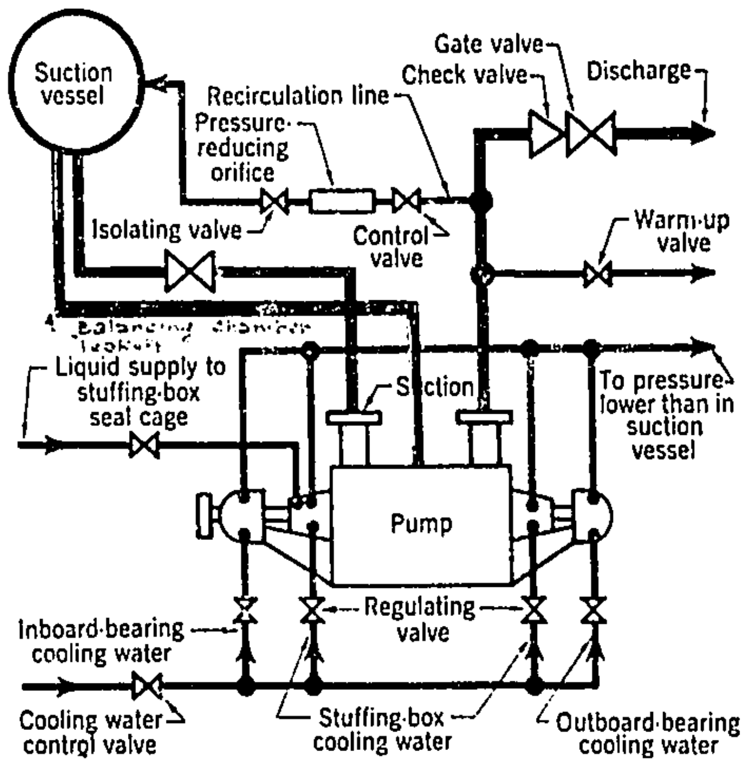


Figure 3.3.1 Various Connections and Auxiliary Services To A Centrifugal Pump

If a pump is not to be used continuously in the desalination process, it should be flushed with clean water after its operation.

3.4 Installation, Foundation and Maintenance Procedures

3.4.1 Installation Procedures

Reciprocating pumps should be set level, accurately aligned with the driver, and located as close as possible to the water supply in order to keep the suction piping short and direct. The required positive suction head should be provided.

The suction piping should be the full size of the pump suction opening and must be absolutely air-tight. If bends are necessary, they should have a long radius. The pump should be equipped with a suction surge chamber and a discharge surge chamber.

Centrifugal pumps should be easily accessible, as close to the feed water supply as possible, and preferably below the level of the liquid to be pumped. Ample space should be provided around the pump for its dismantling and inspection.

Once the baseplate is grouted to the foundation, the installer must check the horizontal, vertical, and angular alignment of the pump and driver shafts. When checking the alignment of "cold" equipment, allowance must be made for the different thermal growth characteristics of the pump and driver. Figure 3.4.1 shows various types of alignment and checking procedures. There are a number of methods for checking alignment; their selection depends upon the type of coupling used. These methods are well described in pumping equipment manuals.

Suction and discharge piping should be of ample size and should be routed directly with a minimum number of bends. Fluid velocity levels of 8 feet per second for suction lines and 15 fps for discharge lines are current practice. The piping should be supported independent of the pump and should be provided with expansion loops for high-pressure service. The piping arrangements must be checked after erection to be sure that piping loads on the pump casing do not cause misalignment of the pump and driver shafts.

The suction pipe must be short, direct, and kept free of air pockets. Long radius elbows preferably should be set vertically. The end of the suction pipe should be the manufacturer's recommended distance below the minimum level of the feed water supply. A pump should never be throttled on the suction side.

The discharge pipe must be equipped with both a check valve and a gate valve. The gate valve is useful in starting and priming the pump, while the check valve will prevent the pump from running backward when stopped.

Auxiliary piping is necessary to provide cooling water for some or all of the following services:

- stuffing-box cooling
- stuffing-box sealing

- lube-oil cooling
- bearing-body cooling
- pump support cooling

By-pass (i.e., recirculating) piping should also be provided to prevent overheating and flashing of the liquid in the pump at low capacities.

3.4.2 Foundation Requirements

Reciprocating pumps should be installed so that they are level and solidly bolted down. Structural steel rails can be used; these are grouted and bolted to the floor. If a poured concrete foundation has been provided, care must be taken to level the unit and rest it on all four feet. For pumps with separate drivers, it is recommended that the pump and the prime mover be mounted on a common base.

Foundations for vertical centrifugal pumps should be rigid and substantial in order to provide a vibration-resistant mounting. The thickness and ground area required for the foundation will depend upon the firmness of the supporting earth. A centrifugal pit is needed for some pumps. The top surface of the concrete foundation should be roughened in order to provide a good bonding surface for the grout.

Foundations for horizontal centrifugal pumps should be sufficiently rigid and substantial to prevent any undue pump vibration and to support the bedplate at all points.

The reinforced concrete foundations should be poured well in advance of pump installation. A template can be used for the foundation bolts. The top surface of the foundation should have a rough finish in order to provide a good bonding surface for the grout. The pump baseplate must be set level before tightening down the foundation bolts; once this last operation is completed, the baseplate should be rechecked with a level and proper adjustment made by wedging or shimming.

Foundation procedures are given in detail in manufacturers' manuals and elevation drawings for each type of pump.

3.4.3 Maintenance Procedures

A list of recommended inspection and maintenance operations is always provided with a new pump. However, some general ground rules as described below are applicable.

1. Reciprocating Pumps

Monthly and semi-annual inspection periods are suggested. However, operating experience acquired with a unit in its particular application will permit a definition of a more flexible schedule. These major parts of the pump must be checked if serious damage is to be avoided:

- valve seats in the cylinder block
- liner seats in the cylinder block
- the lubrication system.

Routine daily checks should be made of the lubricating system oil pressure and oil level, the scavenging system (overflow drain from the spacer block), and pump leakage and noise. Routine checks should be made weekly of the relief valve setting, dippage from plungers, and leakage from between the liner glands and the cylinder block.

Any evidence of pipe scale or foreign matter in the fluid represents a sign of potential damage to all precision fitted parts in the system.

The spare parts which should be kept in stock are: gaskets, seals, valves, and when the pump cannot be shut down for a period, a plunger and liner assembly or possibly a complete set of plungers and liners.

2. Centrifugal Pumps

Hourly or daily inspections should be made for centrifugal pumps. The operator should be alert for irregularities in the operation of the pumps (e.g., unusual noise or vibrations). Any trouble symptoms should be reported and analyzed. Stuffing-box operation and bearing temperature should be checked periodically. An abrupt change in bearing temperature is a sign of trouble.

Semi-annual and annual inspections should be scheduled to check the following:

- check capacity and pressure to determine whether leakage has increased to the point where new wearing-rings, seals, etc. are required.

- check for free movement of the stuffing-box glands, clean and oil the gland bolts and nuts.

- adjust the gland to reduce excessive leakage, and if needed, (leakage is too high) replace the packing.

A complete overhaul of the pump may be required annually depending upon the operating conditions. The pump should be dismantled so that the rotor can be inspected. The condition of the shaft should be checked at the impeller hubs under the shaft sleeves, and at the bearings. There may be signs of leakage along the shaft at the impeller or shaft sleeves (rusting or pitting). Also check for possible shaft distortion due to impeller unbalance. The impellers should be checked for erosion at the inlet. Worn shaft sleeves should be replaced when the stuffing-box packing is changed. The casing waterways must be kept clean and free from rust.

The minimum set of spare parts which should be kept on hand includes:

- one set of bearings

- one set of shaft sleeves and shaft seals

- one complete set of wearing rings (casing and impeller rings)
- one automatic oiler
- material for the casing gasket
- packing for the stuffing-boxes and glands.

If the pump is to be operated continuously and may be subjected to corrosion pitting, a complete rotor should be purchased as a spare with the pump.

3.5 Aeration Problems and Material Selection

Aeration Problems:

The quantity of air admitted to the pump in the feedwater is critical. A small amount of air admitted in the suction has been shown to reduce cavitational pitting in a pump. However, once the acceptable level of air is exceeded, the air forms bubbles in low pressure areas. These bubbles expand and then collapse, causing damage to the material very much like cavitational pitting. The oxygen accompanying the air contributes to corrosion as well.

Some materials withstand the presence of air better than others. A detailed chemical analysis of the feedwater, including its air content, is required before selecting the materials to be used in a pump for saline water service.

Material Selection:

A common pump material is cast iron. It must be covered by a protective paint coating or it will not withstand the corrosive effects of brackish and sea water. Cast iron cannot be used for pump surfaces exposed to high velocities which will erode the paint layer.

Bronze alloys are not suitable for use with the high pressures and velocities of the saline water in the pump waterways.

A report presented by the International Nickel Company, Incorporated, to the Office of Saline Water in 1965 included alloy suggestions for pump components in large multistage flash distillation plants. These pumps operate with low discharge pressures and handle very high flow rates. The recommendations of this report were as follows:

- Pump Shafts: "70 - 30 Ni-Cu Alloy ASTMB127, since this material has greater resistance to pitting and crevice corrosion than Stainless Steel 316, is resistant to stress corrosion cracking, has virtually the same resistance to velocity effects as Stainless Steel 316".

- Pump Casings and Impellers: Stainless Steel 316

- Wearing Rings: "K" Monel

The pump manufacturers contributing to the present study could report very little operating experience with brackish water or sea water for the types of pumps suitable for use in reverse osmosis plants. However, most manufacturers recommend the following materials for the pump components:

- Stainless Steel 316 for all wetted parts

- Ni Resist as an alternative for the casings

- Cast iron with a protective coating was also mentioned for vertical pumps

- "K" Monel and ceramic plungers for reciprocating pumps.

Manufacturer	Pump Type	Speed, rpm	Efficiency, %	Shaft	Class	Material
PLANT NO. 1		HEAD = 867 ft.			CAPACITY = 86.75 gpm	
Worthington	Centrif. - 2 Stages	3550	38.5	Horizontal	1-1/2 HNB-103	Stainless Steel 316
Ingersoll-Rand	Centrif. - 4 Stages	3500	50	Vertical	1-1/2 VP	Stainless Steel 316
Peerless Pumps	Centrif. - 15 Stages	3600	70	Vertical	6 LB-15	Stainless Steel 316
Kobe	Reciprocating	416	90		Size 3F	Stainless Steel 316
Gardner-Denver	Reciprocating	450	90		PS-25	Al-Br + Ceramic Plungers
Byron-Jackson	Centrif. - 12 Stages	3550	68	Vertical	100 VLT	Stainless Steel 316
PLANT NO. 2		HEAD = 867 ft.			CAPACITY = 99.30 gpm	
Worthington	Centrif. - 2 Stages	3550	39.5	Horizontal	1-1/2 HNB-103	Stainless Steel 316
Ingersoll-Rand	Centrif. - 4 Stages	3500	53	Vertical	1-1/2 VP	Stainless Steel 316
Peerless Pumps	Centrif. - 15 Stages	3600	71	Vertical	6 LB-15	Stainless Steel 316
Kobe	Reciprocating	416	90		Size 3F	Stainless Steel 316
Gardner-Denver	Reciprocating	375	90		TA-3	Al-Br + Ceramic Plungers
Byron-Jackson	Centrif. - 12 Stages	3550	70	Vertical	100 VLT	Stainless Steel 316
PLANT NO. 3		HEAD = 867 ft.			CAPACITY = 138 gpm	
Worthington	Centrif. - 2 Stages	3550	45	Horizontal	1-1/2 UNB-11	Stainless Steel 316
Ingersoll-Rand	Centrif. - 6 Stages	3500	54	Vertical	1-1/2 VP	Stainless Steel 316
Peerless Pumps	Centrif. - 15 Stages	3600	78	Vertical	6 LB-15	Stainless Steel 316
Kobe	Reciprocating	371	90		Size 4J	Stainless Steel 316
Gardner-Denver	Reciprocating	500	90		TA-3	Al-Br + Ceramic Plungers
Byron-Jackson	Centrif. - 14 Stages	3550	70	Vertical	200 VLT	Stainless Steel 316
Byron-Jackson	Centrif. - 6 Stages	3550	63	Horizontal	SD 2 x 3 x 7	Stainless Steel 316

Table 3.2.1 Manufacturers' Selections

Manufacturer	Pump Type	Speed, rpm	Efficiency, %	Shaft	Class	Material
PLANT NO. 4		HEAD = 867 ft.			CAPACITY = 867.50 gpm	
Worthington	Centrif. - 2 Stages	3550	68	Horizontal	5 UNB-13	Stainless Steel 316
Ingersoll-Rand	Centrif. - 2 Stages	3500	71	Horizontal	4 GTR	Stainless Steel 316
Peerless Pumps	Centrif. - 11 Stages	1760	82	Vertical	12 LB-11	Stainless Steel 316
Fairbanks-Morse	Centrif. - 2 stages	3500	70	Horizontal	5972	Stainless Steel 316
Byron-Jackson	Centrif. - 11 Stages	1750	80	Vertical	500 VLT	Stainless Steel 316
Byron-Jackson	Centrif. - 2 Stages	3550	74	Horizontal	SD 6 x 8 x 10	Stainless Steel 316
PLANT NO. 5		HEAD = 867 ft.			CAPACITY = 993 gpm	
Worthington	Centrif. - 2 Stages	3550	72	Horizontal	5UNB-13	Stainless Steel 316
Ingersoll-Rand	Centrif. - 2 Stages	3500	73	Horizontal	4 GTR	Stainless Steel 316
Peerless Pumps	Centrif. - 12 Stages	1760	81	Vertical	12 LB-12	Stainless Steel 316
Fairbanks-Morese	Centrif. - 2 Stages	3500	74	Horizontal	5972	Stainless Steel 316
Byron-Jackson	Centrif. - 2 Stages	3550	76	Horizontal	SD 6 x 8 x 10	Stainless Steel 316
PLANT NO. 6		HEAD = 867 ft.			CAPACITY = 1380 gpm	
Worthington	Centrif. - 2 Stages	3550	79	Horizontal	5 UNB-13	Stainless Steel 316
Ingersoll-Rand	Centrif. - 2 Stages	3500	71	Horizontal	4 GTR	Stainless Steel 316
Peerless Pumps	Centrif. - 7 Stages	1760	78	Vertical	15 LC-7	Stainless Steel 316
Fairbanks-Morse	Centrif. - 2 Stages	3500	75	Horizontal	5972	Stainless Steel 316
Byron-Jackson	Centrif. - 2 Stages	3550	74	Horizontal	DVDS 8 x 8 x 12	Stainless Steel 316
Byron-Jackson	Centrif. - 1 Stage	3550	73	Horizontal	SMJ4 x 6 x 15-1/2HH	Stainless Steel 316
PLANT NO. 7		HEAD = 867 ft.			CAPACITY = 8,675 gpm	
Worthington	Centrif. - 2 Stages	1750	81	Horizontal	(2) x 12UZD-1	Stainless Steel 316
Ingersoll-Rand	Centrif. - 2 Stages	1750	85	Horizontal	14 GA	Stainless Steel 316
Peerless Pumps	Centrif. - 7 Stages	1760	82	Vertical	(5) x 15LC-7	Stainless Steel 316
Byron-Jackson	Centrif. - 2 Stages	1780	84	Horizontal	DVSS 14x14x20	Stainless Steel 316

Table 3.2.1 Manufacturers' Selections (Continued)

Manufacturer	Pump Type	Speed, rpm	Efficiency, %	Shaft	Class	Material
PLANT NO. 8		HEAD = 867 ft.		CAPACITY = 9,930 gpm		
Worthington	Centrif. - 2 Stages	1750	84	Horizontal	(2) x 12 UZD-1	Stainless Steel 316
Ingersoll-Rand	Centrif. - 2 Stages	1750	85	Horizontal	14 GA	Stainless Steel 316
Peerless Pumps	Centrif. - ? Stages	1750	82	Vertical	(5) x 15 LC - ?	Stainless Steel 316
PLANT NO. 9		HEAD = 867 ft.		CAPACITY = 13,880 gpm		
Worthington	Centrif. - 2 Stages	1750	83	Horizontal	(2) x 12 UZD-1	Stainless Steel 316
Ingersoll-Rand	Centrif. - 2 Stages	1750	80	Horizontal	14 GA	Stainless Steel 316
Peerless Pumps	Centrif. - ?Stages	1750	82	Vertical	(5) x 15 LC-?	Stainless Steel 316
PLANT NO. 10		HEAD = 1320 ft.		CAPACITY = 86.75 gpm		
Worthington	Centrif. - 5 Stages	3550	46	Horizontal	2 WT-85	Stainless Steel 316
Ingersoll-Rand	Centrif. - 6 Stages	3500	51	Vertical	1-1/2 VP	Stainless Steel 316
Peerless Pumps	Centrif. - 22 Stages	3600	67	Vertical	6 LB-22	Stainless Steel 316
Kobe	Reciprocating	416	90		Size 3F	Stainless Steel 316
Gardner-Denver	Reciprocating	300	90		TA-3	Al-Br, Ceramic Plungers
Gardner-Denver	Reciprocating	175	90		TA-4	Al-Br, Ceramic Plungers
Byron-Jackson	Centrif. - 19 Stages	3550	68	Vertical	100 VLT	Stainless Steel 316
PLANT NO. 11		HEAD = 1320 ft.		CAPACITY = 99.30 gpm		
Worthington	Centrif. - 5 Stages	3550	51	Horizontal	2 WT-85	Stainless Steel 316
Ingersoll-Rand	Centrif. - 7 Stages	3500	53	Vertical	1-1/2 VP	Stainless Steel 316
Peerless Pumps	Centrif. - 22 Stages	3600	67	Vertical	6 LB-22	Stainless Steel 316
Kobe	Reciprocating	416	90		Size 3F	Stainless Steel 316
Gardner-Denver	Reciprocating	350	90		TA-3	Al-Br, Ceramic Plungers
Gardner-Denver	Reciprocating	200	90		TA-4	Al-Br, Ceramic Plungers
Byron-Jackson	Centrif. - 19 Stages	3550	70	Vertical	100 VLT	Stainless Steel 316

Table 3.2.1 Manufacturers' Selections (Continued)

Manufacturer	Pump Type	Speed, rpm	Efficiency, %	Shaft	Class	Material
PLANT NO. 12		HEAD = 1320 ft.			CAPACITY = 138.80 gpm	
Worthington	Centrif. - 6 Stages	3550	58	Horizontal	2 WT-86	Stainless Steel 316
Ingersoll-Rand	Centrif. - 7 Stages	3500	56	Vertical	1-1/2 VP	Stainless Steel 316
Peerless Pumps	Centrif. - 19 Stages	3600	70	Vertical	6 LB-19	Stainless Steel 316
Kobe	Reciprocating	371	90		Size 4J	Stainless Steel 316
Gardner-Denver	Reciprocating	500	90		TA-3	Al-Br, Ceramic Plungers
Gardner-Denver	Reciprocating	250 - 275	90		TA-4	Al-Br, Ceramic Plungers
Byron-Jackson	Centrif. - 22 Stages	3550	70	Vertical	200 VLT	Stainless Steel 316
Byron-Jackson	Centrif. - 10 Stages	3550	70	Vertical	Hydropress 12H	Stainless Steel 316
PLANT NO. 13		HEAD = 1320 ft.			CAPACITY = 867.5 gpm	
Worthington	Centrif. - 4 Stages	3550	71	Horizontal	4 UNQ-11	Stainless Steel 316
Ingersoll-Rand	Centrif. - 8 Stages	3500	72	Horizontal	3 CNTA	Stainless Steel 316
Peerless Pumps	Centrif. - 11 Stages	3600	78	Vertical	10 LA-11	Stainless Steel 316
Byron-Jackson	Centrif. - 4 Stages	3550	80	Horizontal	DVMX 4 x 6 x 9C (2603-2)	Stainless Steel 316
PLANT NO. 14		HEAD = 1320 ft.			CAPACITY = 993 gpm	
Worthington	Centrif. - 4 Stages	3550	72	Horizontal	4 WTF-124	Stainless Steel 316
Ingersoll-Rand	Centrif. - 4 Stages	3500	78	Horizontal	5 HMTA	Stainless Steel 316
Peerless Pumps	Centrif. - 17 Stages	3600	75	Vertical	10 LA-12	Stainless Steel 316
Byron-Jackson	Centrif. - 4 Stages	3550	78	Horizontal	DVMX 4 x 6 x 9C (2603-2)	Stainless Steel 316
PLANT NO. 15		HEAD = 1320 ft.			CAPACITY = 1388 gpm	
Worthington	Centrif. - 3 Stages	3550	78	Horizontal	6WTL-123	Stainless Steel 316
Ingersoll-Rand	Centrif. - 6 Stages	3500	70	Horizontal	5 HMTA	Stainless Steel 316
Peerless Pumps	Centrif. - 11 Stages	1760	78	Vertical	15 LC-11	Stainless Steel 316
Worthington	Centrif. - 2 Stages	3550	68	Horizontal	6 UZD-1	Stainless Steel 316
Byron Jackson	Centrif. - 4 Stages	3550	73	Horizontal	DVMX 4 x 6x 9C (2603-2)	Stainless Steel 316

Table 3.2.1 Manufacturers' Selections (Continued)

Manufacturer	Pump Type	Speed, rpm	Efficiency, %	Shaft	Class	Material
PLANT NO. 16		HEAD = 1320 ft.			CAPACITY = 8,675 gpm	
Worthington	Centrif. - 4 Stages	3550	78	Horizontal	(4) x 6 WTF-124	Stainless Steel 316
Ingersoll-Rand	Centrif. - 2 Stages	1750	85	Horizontal	14 GA	Stainless Steel 316
Peerless Pumps	Centrif. - 11 Stages	1760	82	Vertical	(5) x 15 LC-11	Stainless Steel 316
PLANT NO. 17		HEAD = 1320 ft.			CAPACITY = 9,930 gpm	
Worthington	Centrif. - 4 Stages	3550	78	Horizontal	(4) x 6 WTF-124	Stainless Steel 316
Ingersoll-Rand	Centrif. - 2 Stages	1850	85	Horizontal	14 GA	Stainless Steel 316
Peerless Pumps	Centrif. - ? Stages	1760	92	Vertical	(5) x 15 LC-?	Stainless Steel 316
PLANT NO. 18		HEAD = 1320 ft.			CAPACITY = 13,880 gpm	
Worthington	Centrif. - 4 Stages	3550	78	Horizontal	(6) x 6 WTF-124)	Stainless Steel 316
Ingersoll-Rand	Centrif. - 2 Stages	2000	80	Horizontal	14 GA	Stainless Steel 316
Peerless Pumps	Centrif. - ? Stages	1760	82	Vertical	(5) x 15 LC-?	Stainless Steel 316
Byron-Jackson	Centrif. - 2 Stages	3600	75	Horizontal	DVDS-14 x 16 x 18	Stainless Steel 316
PLANT NO. 19		HEAD = 1765 ft.			CAPACITY = 86.75 gpm	
Worthington	Centrif. - 7 Stages	3550	47	Horizontal	2 WTF-87	Stainless Steel 316
Ingersoll-Rand	Centrif. - 8 Stages	3500	50	Vertical	1-1/2 VP	Stainless Steel 316
Peerless Pumps	Centrif. - 12 Stages	3600	51	Vertical	VDM	Stainless Steel 316
Kobe	Reciprocating	416	90		Size 3F	Stainless Steel 316
Gardner-Denver	Reciprocating	350 - 400	90		TA-3	Al-Br, Ceramic Plungers
Gardner-Denver	Reciprocating	200 - 225	90		TA-4	Al-Br, Ceramic Plungers

Table 3.2.1 Manufacturers' Selections (Continued)

Manufacturer	Pump Type	Speed, rpm	Efficiency, %	Shaft	Class	Material
PLANT NO. 20		HEAD = 1765 ft.			CAPACITY = 99.30 gpm	
Worthington	Centrif. - 7 Stages	3550	51	Horizontal	2 WTF-87	Stainless Steel 316
Ingersoll-Rand	Centrif. - 8 Stages	3500	53	Vertical	1-1/2 VP	Stainless Steel 316
Peerless Pumps	Centrif. - 12 Stages	3600	53	Vertical	VDM	Stainless Steel 316
Kobe	Reciprocating	416	90		Size 3F	Stainless Steel 316
Gardner-Denver	Reciprocating	450	90		TA-3	Al-Br, Ceramic Plungers
Gardner-Denver	Reciprocating	250 - 300	90		TA-4	Al-Br, Ceramic Plungers
PLANT NO. 21		HEAD = 1765 ft.			CAPACITY = 138.8 gpm	
Worthington	Centrif. - 8 Stages	3550	58	Horizontal	2 WTF-88	Stainless Steel 316
Ingersoll-Rand	Centrif. - 8 Stages	3500	56	Vertical	1-1/2 VP	Stainless Steel 316
Peerless Pumps	Centrif. - 14 Stages	3600	56	Vertical	VDM	Stainless Steel 316
Kobe	Reciprocating	371	90		Size 4J	Stainless Steel 316
Gardner-Denver	Reciprocating	350 - 400	90		TA-4	Al-Br, Ceramic Plungers
Byron-Jackson	Centrif. - 30 Stages	3550	70	Vertical	200 VLT	Stainless Steel 316
PLANT NO. 22		HEAD = 1765 ft.			CAPACITY = 867.5 gpm	
Worthington	Centrif. - 4 Stages	3550	75	Horizontal	4 WTF-124	Stainless Steel 316
Ingersoll-Rand	Centrif. - 7 Stages	3500	74	Horizontal	4 HMTA	Stainless Steel 316
Byron-Jackson	Centrif. - 9 Stages	3550	75	Vertical	400 VLT	Stainless Steel 316
Byron-Jackson	Centrif. - 5 Stages	3550	80	Horizontal	DVMX 4 x 6 x 9C (2603-2)	Stainless Steel 316
PLANT NO. 23		HEAD = 1765 ft.			CAPACITY = 993 gpm	
Worthington	Centrif. - 4 Stages	3550	76	Horizontal	4 WTF-124	Stainless Steel 316
Ingersoll-Rand	Centrif. - 5 Stages	3500	78	Horizontal	5 HMTA	Stainless Steel 316
Byron-Jackson	Centrif. - 4 Stages	3550	79	Horizontal	DVMX 4 x 6 x 9D	Stainless Steel 316

Table 3.2.1 Manufacturers' Selections (Continued)

Manufacturer	Pump Type	Speed, rpm	Efficiency, %	Shaft	Class	Material
PLANT NO. 24	HEAD = 1765 ft.				CAPACITY = 1388 gpm	
Worthington	Centrif. - 4 Stages	3550	78	Horizontal	6 WTF-124	Stainless Steel 316
Ingersoll-Rand	Centrif. - 5 Stages	3500	78	Horizontal	6 HMTA (low pres. casing)	Stainless Steel 316
Byron-Jackson	Centrif. - 4 Stages	3550	80	Horizontal	DVMX 4 x 6 x 9D	Stainless Steel 316
Byron-Jackson	Centrif. - 5 Stages	3550	78	Horizontal	DVMX 4 x 6 x 10B	Stainless Steel 316
PLANT NO. 25	HEAD = 1765 ft.				CAPACITY = 8,675 gpm	
Worthington	Centrif. - 4 Stages	3550	81	Horizontal	(4) x 6 WTF-124	Stainless Steel 316
Ingersoll-Rand	Centrif. - 4 Stages	3500	80	Horizontal	(4) x 8 HMTA	Stainless Steel 316
Byron-Jackson	Centrif. - 2 Stages	3600	84.5	Horizontal	DVDS 14 x 16 x 18	Stainless Steel 316
PLANT NO. 26	HEAD = 1765 ft.				CAPACITY = 9,930 gpm	
Worthington	Centrif. - 5 Stages	3550	79	Horizontal	(4) x 6 WTF-125	Stainless Steel 316
Ingersoll-Rand	Centrif. - 4 Stages	3500	79	Horizontal	(4) x 8 HMTA	Stainless Steel 316
Byron-Jackson	Centrif. - 2 Stages	3600	85	Horizontal	DVDS 14 x 16 x 18	Stainless Steel 316
PLANT NO. 27	HEAD = 1765 ft.				CAPACITY = 13,880 gpm	
Worthington	Centrif. - 5 Stages	3550	82	Horizontal	(6) x 6 WTF-125	Stainless Steel 316
Ingersoll-Rand	Centrif. - 4 Stages	3500	79	Horizontal	(6) x 8 HMTA	Stainless Steel 316
Byron-Jackson	Centrif. - 2 Stages	3600	82.5	Horizontal	DVDS 14 x 16 x 11	Stainless Steel 316
PLANT NO. 28	HEAD = 3340 ft.				CAPACITY = 173.30 gpm	
Worthington	Centrif. - 9 Stages	3550	58	Horizontal	2-1/2 WTZ-129	Stainless Steel 316
Ingersoll-Rand	Centrif. - 17 Stages	3500	65	Vertical	2-1/2 VHTB	Stainless Steel 316
Ingersoll-Rand	Centrif. - 9 Stages	3500	52	Horizontal	3 HMTA	Stainless Steel 316
Gardner-Denver	Reciprocating	175	85		PA-8	Al-Br,Ceramic Plungers
Byron-Jackson	Centrif. - 12 Stages	3550	57	Horizontal	DVMX 3 x 4 x 9 A	Stainless Steel 316

Table 3.2.1 Manufacturers' Selections (Continued)

Manufacturer	Pump Type	Speed, rpm	Efficiency, %	Shaft	Class	Material
<u>PLANT NO. 29</u>		HEAD = 3340 ft.			CAPACITY = 1,733 gpm	
Worthington	Centrif. - 7 Stages	3550	77	Horizontal	6 WTF-127	Stainless Steel 316
Ingersoll-Rand	Centrif. - 7 Stages	3500	78	Horizontal	6 HMTA (high pres. casing)	Stainless Steel 316
Byron-Jackson	Centrif. - 7 Stages	3550	81	Horizontal	DVMX 4 x 6 x 10C (2701-1)	Stainless Steel 316
<u>PLANT NO. 30</u>		HEAD = 3340 ft.			CAPACITY = 17,330 gpm	
Worthington	Centrif. - 9 Stages	3550	80	Horizontal	(7) x 6 WTF-129	Stainless Steel 316

Table 3.2.1 Manufacturers' Selections (Continued)

NPSH Requirements (min)

Worthington	-	horizontal	40 feet
Ingersoll-Rand	-	vertical horizontal	40 feet 40 feet
Peerless Pumps	-	vertical	50 feet
Byron-Jackson	-	vertical horizontal	20 feet 30 feet
Fairbanks Morse	-	horizontal	30 feet
Kobe	-	reciprocating	100 feet
Gardner-Denver	-	reciprocating	20 feet

<u>Note</u>

In Class column whenever (5) x class is indicated, it means that 5 units of that class are operated in parallel

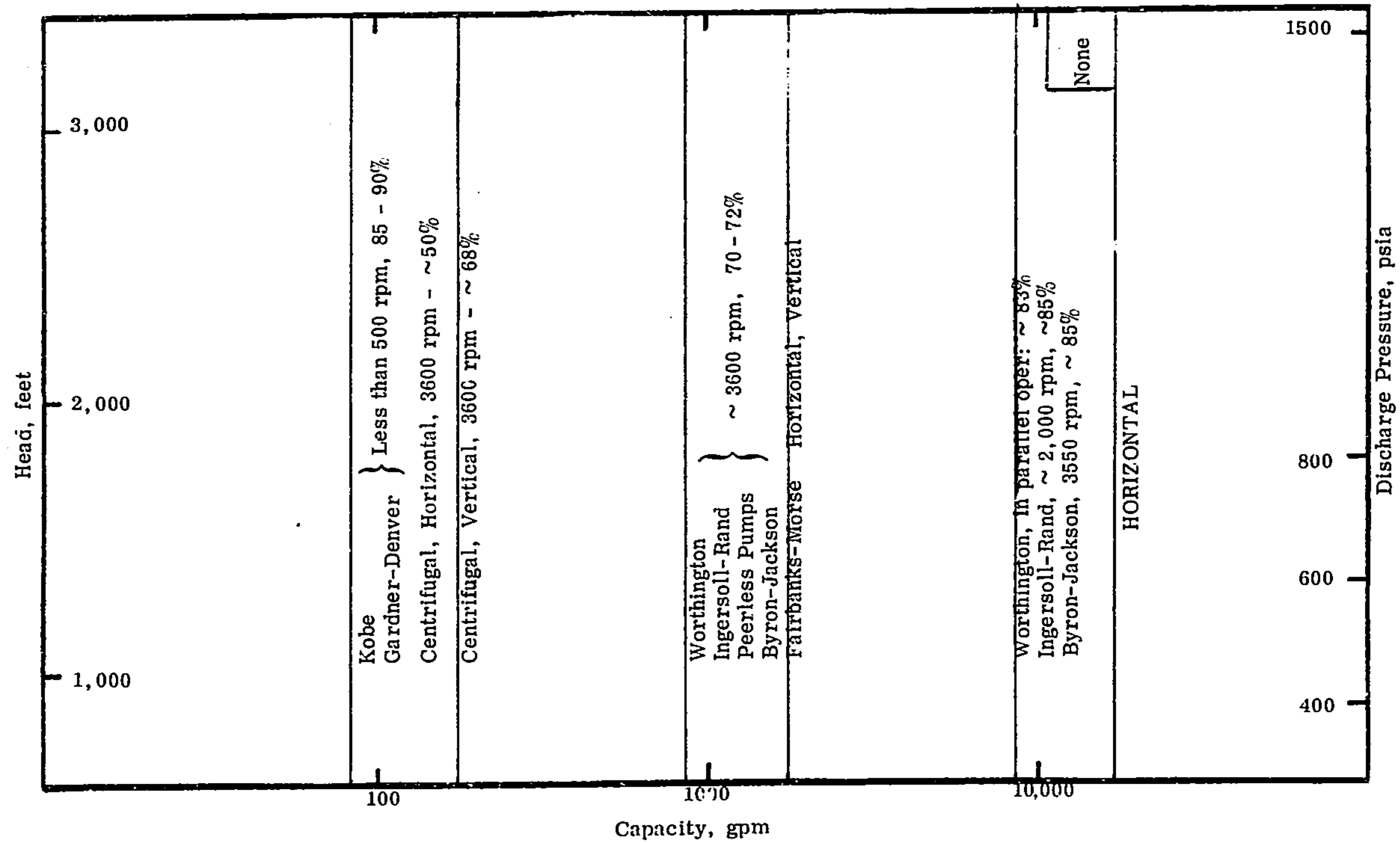
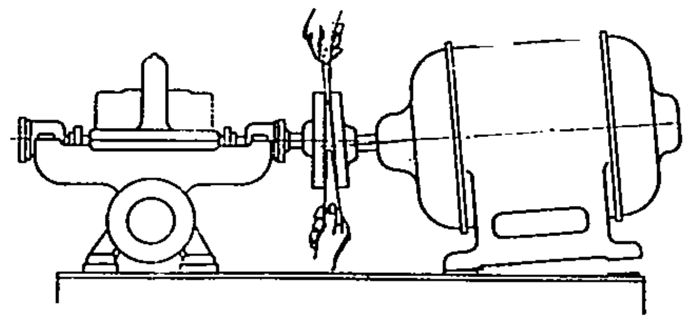
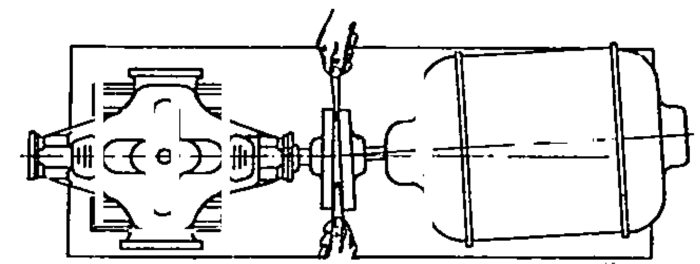


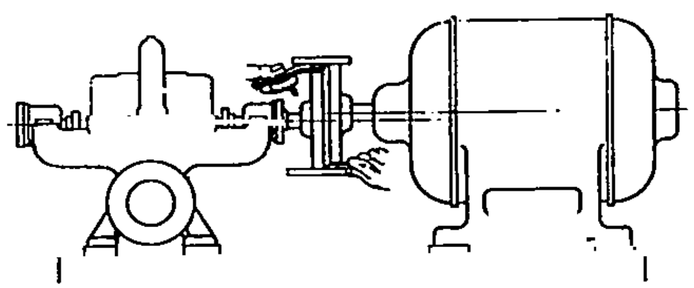
Figure 3.2.1 Pumps Selected for the Various Plant Design Requirements of Head and Capacity



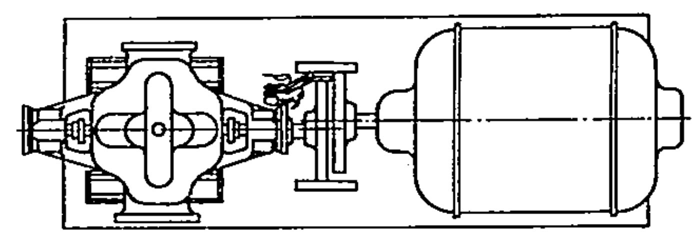
Checking Vertical Angular Misalignment



Checking Horizontal Angular Misalignment



Checking Vertical Alignment



Checking Horizontal Alignment

Figure 3.4.1 Various Alignment Checking Techniques
(From Ingersoll-Rand)

Section 4

SELECTION OF ENERGY RECOVERY SYSTEMS

4.1 Introduction and Approach

A preliminary evaluation of hydraulic turbines for use in energy recovery systems was presented in Section 2.4. As a result of this evaluation, Pelton impulse turbines and Francis reaction turbines were selected as being suitable for reverse osmosis desalination plants. The design requirements for these turbines, corresponding to the "standard" plants of Section 2.1 including the effects of brine-side pressure drop, are given in Figure 4.1.1. These design requirements were incorporated in a questionnaire sent to hydraulic turbine manufacturers. The questionnaire, which is included in this report as Appendix B, requested detailed technical pricing information on currently-offered equipment.

4.2 Manufacturers Contacted

The hydraulic turbine questionnaire of Appendix B was sent to the manufacturers listed in Table 4.2.1. The list of hydraulic turbine manufacturers is much shorter than the list of pump manufacturers. Only a few of the manufacturers responded to the inquiry. Hydraulic turbines are custom-designed machines, so it is more difficult for the turbine manufacturers to reply to the questionnaire than it was for the pump manufacturers with their standard product lines. The turbine manufacturers who did respond and a list of their selections are presented in Table 4.2.2. Most of the hydraulic turbines selected have efficiencies of 85% or higher.

4.3 Operation of Pelton Turbines

The needle position in the Pelton turbine nozzle can be controlled automatically or manually. The needle position can be used to regulate the pressure and flow rate produced by the pump and admitted to the membrane system. Thus, the needle valve can take the place of the throttle valve arrangement which is required to regulate the brine-side membrane pressure in the reverse osmosis process without energy recovery.

The manufacturers believe that speed governors will not be necessary for turbines in this application. However, this point should be reviewed for specific plant arrangements when the method of loading the turbine (direct pump drive vs. electrical generator) is defined and the control scheme for the plant has been selected.

During shutdown of the plant, or shutdown of the turbine alone, the turbine must be slowed down gradually to avoid generating water hammer which could damage the membranes.

4.4 Installation, Foundation and Maintenance Requirements

The installation and maintenance requirements for Pelton turbines vary slightly from manufacturer to manufacturer. However, general ground rules can be established for most units without special regard to their size.

4.4.1 Installation Requirements

The inlet piping to the hydraulic turbine should have a minimum number of elbows in order to reduce both head loss and turbulence. Precise alignment of the inlet nozzle and runner is necessary in order to obtain maximum efficiency and minimize the side thrust. The installation of a Pelton turbine must allow for formation of a free water surface below the wheel.

For Francis turbines, it may be necessary under certain operating conditions to provide a positive discharge head to prevent cavitation. Pelton turbines can discharge into a concrete sump, whereas Francis turbines require the design and construction of a draft tube.

4.4.2 Foundation Requirements

Hydraulic turbines should be solidly anchored on concrete foundations. Depending on size, the machines may be equipped with a baseplate or several sole plates which should be aligned and anchored to the foundation.

4.4.3 Maintenance Requirements

The erosive and corrosive action of the brine on the needle, nozzle, and runner buckets of Pelton turbines will dictate the maintenance requirements for these units.

Wear can seriously affect the efficiency of these machines. After the initial start-up, monthly inspections should be made. These inspections should include usual observations of wear. After operating experience has been acquired, it may be possible to reduce the frequency of the inspection periods to semi-annual or annual.

Daily maintenance tasks include checking the oil level in the lubricating system and monitoring the bearing temperature level.

Spare parts which should be stocked are a replacement runner, needle, needle actuator, and seals or packing. Additional spares may be found to be necessary after operating experience provides data on the erosion characteristics of the materials used with the concentrated brine. The seals on the shaft driving the generator or pump should be checked for excessive leakage.

4.5 Material Selection

The salt concentration which the turbines will encounter in both brackish water and sea water service was given to the manufacturers in the questionnaire. Pelton turbines are not often used with saline water at present. Therefore, material requirements have not been well defined for these machines. The high velocity of impingement of the jet upon the bucket is likely to require the use of high quality stainless steel (e.g., 316 Stainless) for the needle and runner. As an alternative, 13% chrome cast steel (Type 420) was also suggested for the runners and nozzles. The turbine casing can be made of cast iron with a protective coating over the interior surfaces.

Manufacturers	Answer
Toshiba Machine Co., Ltd. Fuji Building 3, 4-Chome Ginza Nishi Chuo-Ku Tokyo, JAPAN	No information.
Mr. J. H Lieber Director of Siaes Charmilles Engineering Works Ltd. 109, rue de Lyon Geneva, Switzerland	Information not received.
Mr. Andre Puyo Directeur Division Energie Establissements NEYRPIC Avenue de Beauvert Grenoble, FRANCE	No information.
Costruzioni Meccaniche Riva - Calzoni S.p.A. Via Stendhal, 34 Milan, ITALY	No information.
Franco-Tosi San Giorgio S.A.I. Corso Italia, 27 Legnano, Milan, ITALY	No information.
J. M. Voith, G.m.b.H., Heidenheim (Brenz) GERMANY	Information provided.
Mr. G. V. Arata Manager, Utility Sales Baldwin-Lima-Hamilton Industrial Equipment Division Philadelphia, Pa. 19142	No information
Mr. H. A. Mayo, Jr. Field Sales Manager Allis-Chalmers York Plant, Hydraulic Products Division York, Pennsylvania 17405	Information provided.
Mr. J. Sawyer General Turbine Corporation 1559 Niagara Street Buffalo, New York 14313	No information.

Table 4.2.1 List of Hydraulic Turbine Manufacturers

Manufacturers	Answer
Escher Wyss G. m. b. H. Ravensburg, WEST GERMANY	Information provided
Mr. J. Robert Groff President and General Manager James Leffel & Co. Springfield, Ohio	Information provided.

Table 4. 2. 1 List of Hydraulic Turbine Manufacturers (Continued)

Manufacturer	Type	No. Jets	Eff.	Speed (rpm)
James Leffel & Co.	Pelton	1	85%	3600 1800 1200 900
Allis-Chalmers	Pelton	1	87 - 89%	3600
Voith GMBH	Pelton	1	85 - 87%	3600 1800
		2	85 - 87%	3600 1800
	Francis		85%	3600 1800
Escher-Wyss GMBH	Pelton	1	80 - 88% 83 - 85%	3600 1800
		2	86%	3600
		4	87.5%	3600

Table 4 2.2 Hydraulic Turbines Offered by the Manufacturers

Pump Head	% Rec.	% Head Loss	10^5 GPD 0	10^5 GPD 5	10^5 GPD 10	10^5 GPD 15	10^6 GPD 0	10^6 GPD 5	10^6 GPD 10	10^6 GPD 15	10^7 GPD 0	10^7 GPD 5	10^7 GPD 10	10^7 GPD 15
867 FT.	50	FT	867	822	777	732	867	822	777	732	867	822	777	732
		GPM	17.3			→	173.			→	1730			→
		HP	3.2	3.05	2.9	2.7	32.	30.5	29.	27	320	305	290	270
	70	FT	867	822	777	732	867	822	777	732	867	822	777	732
		GPM	29.9			→	299.			→	2990			→
		HP	5.5	5.3	4.9	4.7	55.	53.	49.	47.	550	530	490	470
	80	FT	867	822	777	732	867	822	777	732	867	822	777	732
		GPM	69.4			→	694.			→	6940			→
		HP	12.9	12.2	11.5	10.9	129	122	115	109	1290	1220	1150	1090
1320 FT.	50	FT	1320	1250	1180	1115	1320	1250	1180	1115	1320	1250	1180	1115
		GPM	17.3			→	173.			→	1730			→
		HP	4.9	4.6	4.4	4.1	49.	46.	44	41	490	460	440	410
	70	FT	1320	1250	1180	1115	1320	1250	1180	1115	1320	1250	1180	1115
		GPM	29.9			→	299.			→	2990			→
		HP	8.5	8.0	7.6	7.1	85.	80	76	71	850	800	760	710
	80	FT	1320	1250	1180	1115	1320	1250	1180	1115	1320	1250	1180	1115
		GPM	69.4			→	694.			→	6940			→
		HP	19.6	18.6	17.6	16.6	196.	186	176	166	1960	1860	1760	1660
1765 FT.	50	FT	1765	1675	1590	1495	1765	1675	1590	1495	1765	1675	1590	1495
		GPM	17.3			→	173.			→	1730			→
		HP	6.5	6.2	5.9	5.5	65.	62.	59.	55.	650	620	590	550
	70	FT	1765	1675	1590	1495	1765	1675	1590	1495	1765	1675	1590	1495
		GPM	29.9			→	299.			→	2990			→
		HP	11.3	10.7	10.2	9.6	113.	107.	102.	96.	1130	1070	1020	960
	80	FT	1765	1675	1590	1495	1765	1675	1590	1495	1765	1675	1590	1495
		GPM	69.4			→	694.			→	6940			→
		HP	26.3	24.9	23.7	22.3	263	249	237	223	2630	2490	2370	2230
3340 FT.	40	FT	3340	3175	3000	2835	3340	3175	3000	2835	3340	3175	3000	2835
		GPM	104.			→	1040.			→	10,400			→
		HP	74.5	70.9	67.	63.2	745	709	670	632	7450	7090	6700	6320

Figure 4.1.1 Hydraulic Turbine Design Requirements

Section 5

SELECTION OF DRIVING SYSTEMS AND GEARS

5.1 Available Power Supplies

Three forms of power supply have been selected as alternatives for the present study of reverse osmosis desalination plants. These are:

- electricity
- steam
- diesel fuel

In Section 2.3 of this report, background information was provided on electric motors, steam turbines, and diesel engines for use in pumping systems. After preliminary selections of suitable types of drivers were made as discussed in that section, the appropriate manufacturers were contacted for additional information. Final selections of drivers for reverse osmosis pumps, based upon the manufacturers' recommendations, are discussed in the sections which follow.

5.2 Electric Motor Selection

A general description of the various electric motor types was given in Section 2.3.2.

A number of electric motor manufacturers were contacted for technical and cost information on squirrel-cage induction motors. The manufacturers contacted were:

Westinghouse Electric Corporation

General Electric Company

Allis-Chaimers Company

Peerless Electric Company

Table 5.2.1, below, lists the usual voltage ratings of these alternating current motors.

Horsepower Range	Voltage Rating
0 - 200	220, 440 or 550
200 - 500	440, 550 or 2300
500 - 1000	440, 550 or 2300
1000 - 5000	2300
5000 & above	2300, 4000 or 4160

Table 5.2.1 Voltage Rating of Squirrel-Cage Induction Motors

Most of these motors are standard items. However, for some of the large sizes above 5,000 HP, motors are custom designed for the specific application. Their design takes into consideration the driven equipment characteristics such as starting torque, full load horsepower, and type of duty expected.

Various types of enclosures (e.g., dripproof, explosion-proof, totally enclosed, open, etc.) can be provided for these motors. The enclosures affect the cost of the motor according to the kind of protection that they provide. Complete descriptions of these enclosures are given in motor sales catalogs and operating manuals.

It is desirable to start induction motors by making a direct connection across the power line. However, in the larger ratings (larger than 200 HP), it may be necessary to employ reduced-voltage starting in order to meet starting-current restrictions if the required starting torque is high.

A compensator or autostarter can be used for reduced-voltage starting. Another method used to start squirrel-cage induction motors consists of resistors which are inserted in series with the stator and are gradually cut out as the motor comes up to speed. Resistor starters are less expensive than autostarters, but in the case of a slow start the resistors may burn out.

The motor can be mounted on the base-plate together with the pump, or installed close to the pump by appropriate gearing or coupling. The motor shaft could be carefully aligned with the pump shaft.

Electric motors require very little maintenance and service. They are extremely reliable machines. Motors are protected against burnout by thermal overload relays. During operation, the only service usually required is routine checking of kilowatt input, speed, and bearing temperatures.

5.3 Steam Turbine Selection

The types of steam turbines considered practical for use to drive the pumps in a reverse osmosis plant are the straight-through condensing or non-condensing turbines. Special characteristics of these machines are described in Section 7.4.2, since these characteristics are related to the turbine cost and steam consumption.

The manufacturers producing steam turbines of sufficient capacity are:

Worthington Corporation

Westinghouse Electric Corporation

General Electric Company

Allis-Chalmers Company

The service requirements for steam turbines are not demanding. Once or twice a week, the lube oil level and pressure should be checked. The operator should be alert to changes in sound or vibration characteristics which may indicate trouble in the machine. Steam conditions should be checked routinely.

5.4 Diesel Engine Selection

Manufacturers who supplied information on diesel engines for application in our pumping systems were as follows:

Fairbanks-Morse Power Systems Division of Colt Industries

Worthington Corporation

General Motors Company

Cummins Engine Company

Detroit Diesel Company

The overall efficiencies for the engines offered ranged from 28% for small engine sizes up to 40% for the largest engine sizes.

Diesel engine speed is generally related to the delivered power. A listing of standard values of power level versus speed for commercially available diesel engines is provided in Table 5.4.1.

Horsepower Range (HP)	Typical Shaft Speed Available (rpm)
100 - 320	1800
320 - 900	1200
900 - 3000	720
3000 - 18000	400

Table 5.4.1 Typical Diesel Engine Speed Vs. Power Characteristics

Long service life can be expected from diesel engines. Maintenance requirements are minor, although piston rings may have to be replaced after about 8000 hours of operation.

These engines are generally equipped with a pressure lubrication system which supplies a continuous flow of oil to all surfaces requiring lubrication and also to the pistons for cooling. Thus lube oil system must be checked and the oil supply in the header must be kept at the proper level.

5.5 Gears

Gears are required in some of the proposed pumping system arrangements to transmit power with a concomitant change in shaft speed. Tables 5.5.1 and 5.5.2 show the range of combinations of speed, power, and speed ratio which are required for the pumping systems considered in this study.

Information on pricing and availability of the gears shown in these tables was requested and obtained from the following manufacturers:

Falk Corporation

Philadelphia Gear Corporation

Western Gear Corporation

For most of the gears in the tables, the power transmission efficiencies exceed 95%.

Horsepower (HP)	Gear Input Speed (rpm)	Speed Ratio
20 to 20,000	400	2:1 to 10:1
20 to 20,000	1200	2:1 to 10:1
20 to 20,000	3600	2:1 to 10:1

Table 5.5.1 Speed Reducing Gears Considered for Use in the Reverse Osmosis Plants of This Study

Horsepower (HP)	Gear Input Speed	Speed Ratio
100 to 10,000	400	3:1 to 9:1
100 to 10,000	1200	3:1 to 6:1
100 to 10,000	3600	3:1 to 5:1

Table 5.5.2 Speed Increasing Gears Considered for Use in the Reverse Osmosis Plants of This Study

Section 6

TOTAL SYSTEM CONFIGURATION

6.1 General Plant Layout

The preceding sections discussed the selection of the equipment needed for pumping feed-water and recovering energy from the rejected brine in reverse osmosis desalination plants. In this Section 6, alternative combinations of the drivers, pumps, and energy recovery schemes will be described.

A general plant layout (Figure 6.1.1) shows most of the components that are of major importance in a reverse osmosis desalination plant.

6.1.1 Matching of Various Components

Two major systems can be independently defined and then matched:

- A "driving system" which is comprised of the driver, its power supply, their associated controls, and the gear or coupling to the pump.
- A "water system" which includes the pump, the membrane arrays and the energy recovery system.

Figure 6.1.2 describes the matching problem in schematic fashion. The driving system has its own efficiency and provides an output shaft speed and horsepower level. The output from the driving system must match the input required by the pump. The input horsepower and speed for the pump depends upon the feedwater flow rate and pressure level requirements and the pressure losses encountered in the different piping sections shown in Figure 6.1.1.

6.1.2 Alternative Energy Recovery Systems

The brine rejected at high pressure from the membrane channels can be supplied to a Pelton turbine. The turbine discharges the brine at atmospheric pressure into a sump and develops shaft power which can be used in either of two ways:

- To supply part of the shaft power to the pump through direct gearing

- To drive an electrical generator which produces electrical power available for sale or use in the plant.

These alternative energy recovery systems, when matched to the pump and its driver, as described in the next section, constitute the various pumping system arrangements that have been investigated in this program.

6.2 Discussion of Various Arrangements

Four types of drivers have been selected:

- squirrel-cage induction motor

- non-condensing steam turbine

- condensing steam turbine

- diesel engine

By combining the four types of drivers with the two different energy recovery systems, eight possible plant arrangements can be devised. These plant arrangements are shown on Figure 6.2.1. This figure shows schematically the eight different driving and energy recovery arrangements which were analyzed for both "brackish water" plants and "sea water" plants in this investigation.

Each of the individual plant arrangements can have within it a number of different shaft speeds for the various components. One of the objectives of this study is to determine the optimum speeds for the various components in terms of capital and operating costs. Computer programs, described in Section 8, were prepared to carry out the cost analysis. The output is presented later in Section 8.2.2.

The eight differing plant arrangements were designated as follows:

Arrangement A - Uses an electric motor to drive the pump. A hydraulic turbine recovers the power available in the brine stream and drives a generator. The electrical power generated is used to provide part of the power for the motor. This arrangement is analyzed in computer Program No. 4.

Arrangement B — Also has a motor which drives the pump. The hydraulic turbine is coupled to the driving shaft of the pump by means of gears, thus providing extra power to permit reduction of the size of the motor (Program No. 1).

Arrangement C — Employs a non-condensing steam-turbine to drive the pump. The hydraulic turbine is coupled to a generator creating electrical power for sale (Program No. 2).

Arrangement D — Similar to B, with the non-condensing steam turbine replacing the electric motor (Program No. 2)

Arrangements E and F — Are the respective counterparts of arrangements C and D. the non-condensing steam-turbine being replaced by a condensing unit (Program No. 3).

Arrangements G and H — Are the respective counterparts of Arrangements E and F, where the condensing steam-turbine has been replaced by a diesel engine. (Program No. 5).

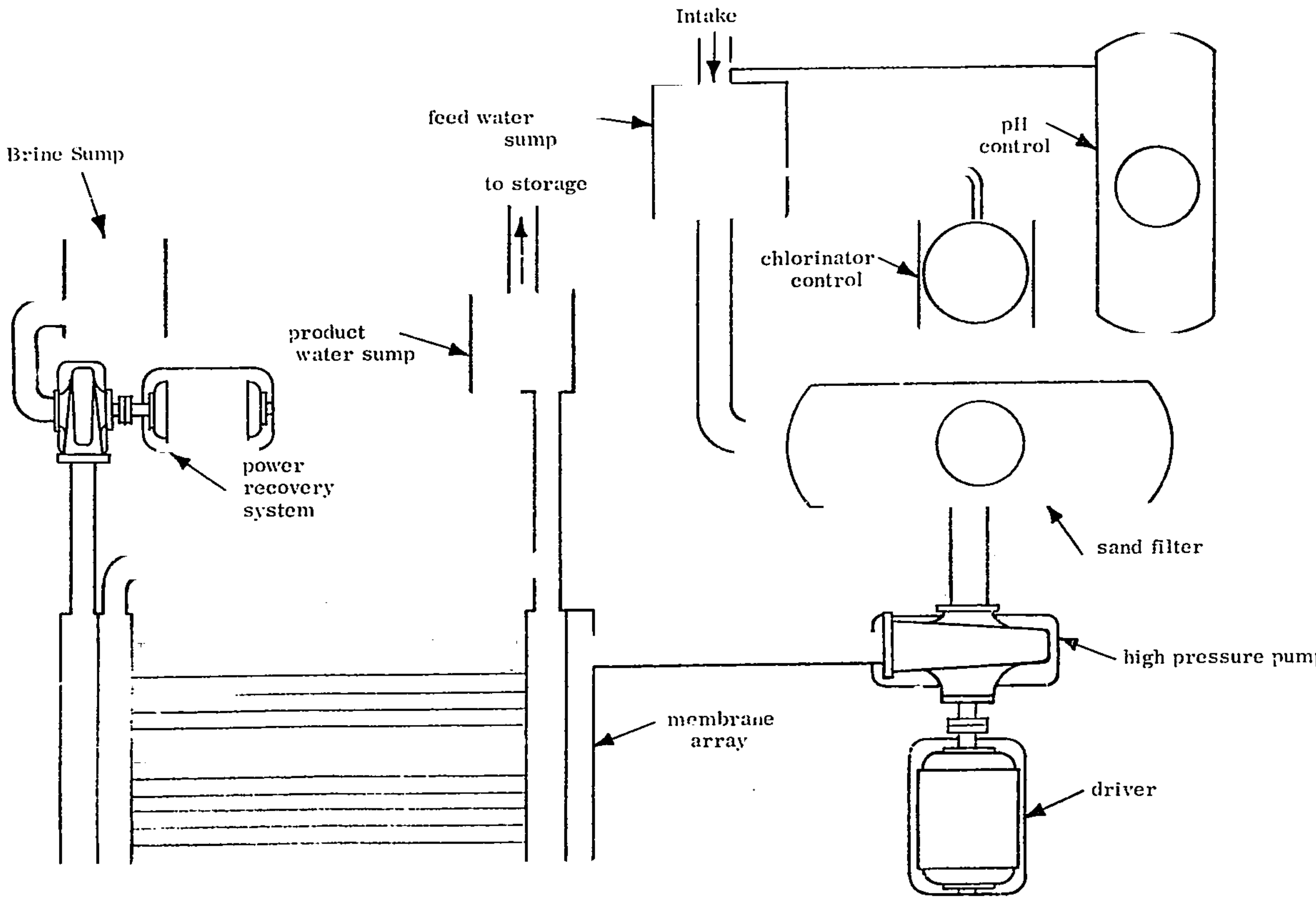


Figure 6. 1. 1 - General Plant Layout

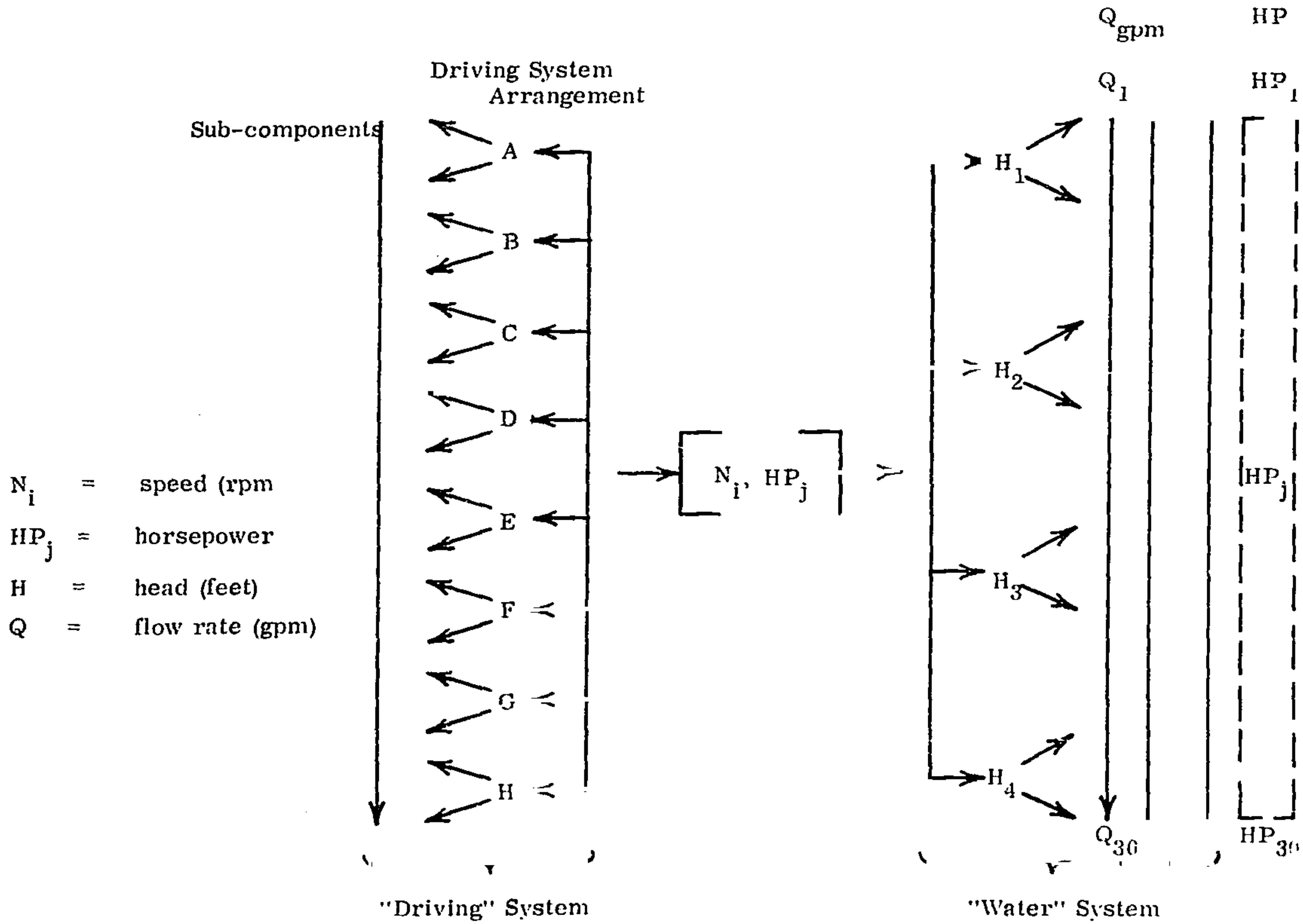


Figure 6.1.2 Network Relating the "Water" System to the "Driving" System

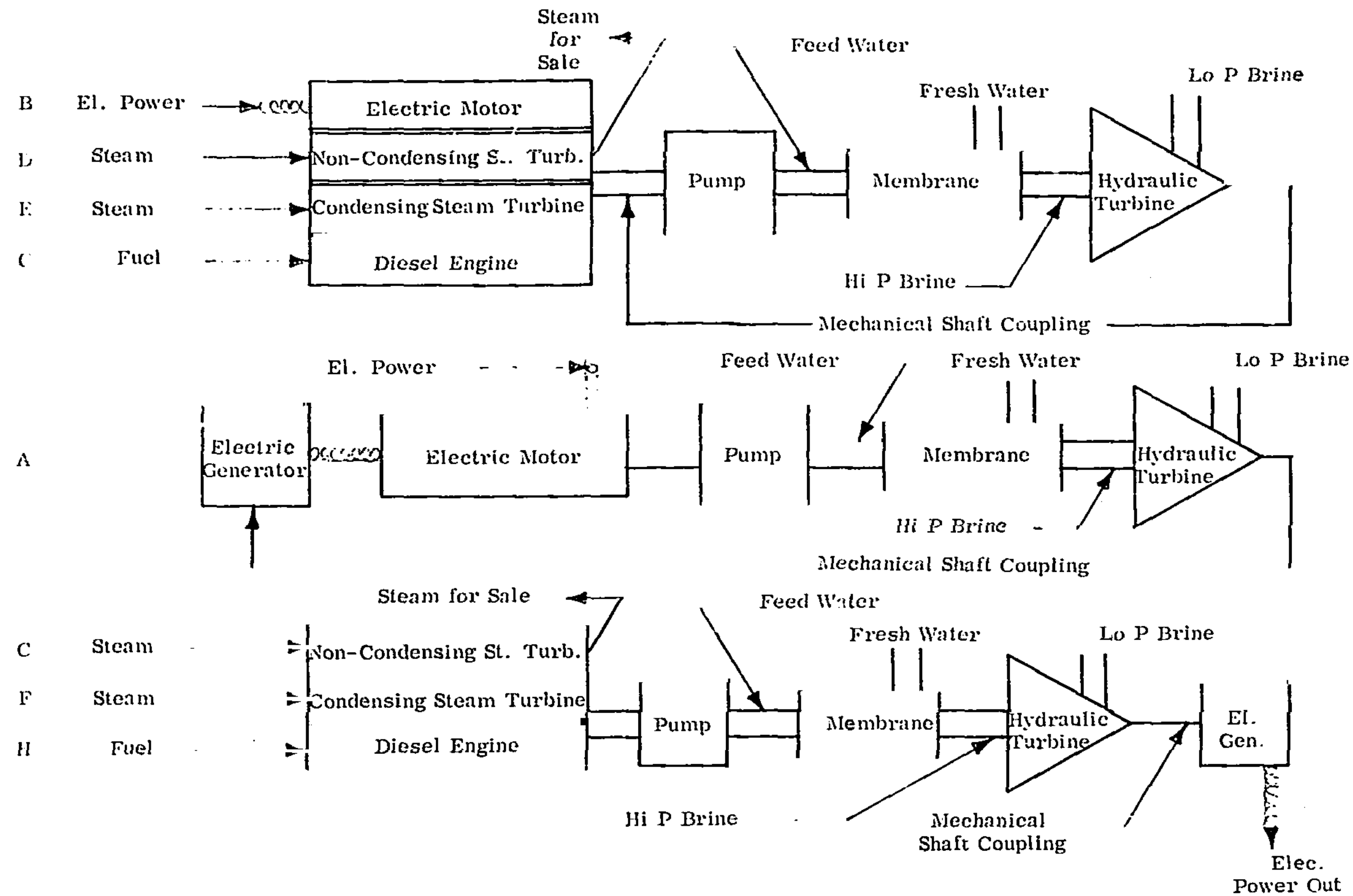


Figure 6.2.1 Various Plant Arrangements

Section 7

COST CORRELATIONS

7.1 Introduction and Purpose

The previous sections have described the technical aspects of the pumping equipment, energy recovery equipment and other pieces of machinery that were part of the overall study. The economic considerations for the pumping system components, e.g., first cost and operating costs, are discussed in this section.

Cost information was gathered from manufacturers' data and transformed for use in computer analysis of the contributions of the pumping systems to the cost of fresh water production.

Service and maintenance cost information was not readily available for pumps and hydraulic turbines from manufacturers' sources. Therefore, Dynatech prepared estimates for these costs as discussed in Section 7.8. These estimates were submitted to pump and hydraulic turbine manufacturers; they agreed that the figures presented were not unreasonable. These service and maintenance cost figures are presented in the computer program as "operating costs" and will be referred to as such.

The price information (i.e., capital costs) presented in this section were estimated by the manufacturers as of May, 1968. These prices are accurate within 10% of the final price of the unit selected.

The cost of power supplied to the pump driver (steam, fuel oil, or electricity), membrane costs and installation costs, and similar desalination plant cost information have been provided for this study by the Office of Saline Water.

7.2 Pump Costs

In order to establish an equation or graphical curve relating pump capital cost to a suitable parameter which describes the corresponding unit (e.g., horsepower, specific speed, flow, etc.), it is necessary to treat each type of pump individually.

Reciprocating pumps can be of the piston type or the plunger type and can have one or more cylinders. The cost of a reciprocating pump depends upon its rating, which is usually its maximum horsepower. The materials used for its construction also influence its cost.

Centrifugal pumps can have a vertical shaft or horizontal shaft and a single-suction or double-suction first stage. The number of stages and the impeller diameter are important factors in establishing their cost. Shaft speeds and the casing construction may differ considerably from manufacturer to manufacturer.

A number of cost correlations have been previously established for specific types of pumps or types of service. In these cost correlations, an index describing the pump (which is generally restricted to single-stage, single suction) is plotted versus the cost of the unit (Ref. 14 and 15). So many restrictions have been imposed on the units selected that the correlating parameters and results are not useful for the present study. Instead, new results and correlating methods had to be developed as described in the sections which follow.

7.2.1 Reciprocating Pumps

Information received from the reciprocating pump manufacturers did not cover a very extensive range of machine types and cost figures. Contributors to the information (Section 3.2) were:

Gardner-Denver Company
Kobe, Incorporated

The cost data was restricted to triplex plunger pumps. Two pump sizes were quoted by Kobe (60 HP and 125 HP) while many sizes were quoted by Gardner-Denver (Table 7.2.1)

Figure 7.2.1 shows a graphical representation of their cost information. The costs plotted included special materials for fluid ends, special alloys for the plungers (or ceramic plunger) a relief valve, and the required V-belt drive and its belt guard. Since the belt drive cost is included in the pump cost, the cost of the gear and belt were set equal to zero in the computer programs when reciprocating pumps were selected.

Manufacturer	Pump Class	Efficiency	Price
Kobe	Size 3F	90%	$ 7,000
	Size 4J	90%	$13,000
Gardner-Denver	PS-25	90%	$ 2,300
	TA-3	90%	2,800
	TA-4	90%	4,800
	TA-5	85%	9,000
	PA-8	85%	16,000

Table 7.2.1 Reciprocating Pump Prices and Efficiencies

Pump Class	Efficiency %	Plant No.	Price
1-1/2 HNB-103	38.5 39.5	1, 2	$ 4,700
1-1/2UNB-11	45	3	4,500
5 UNB-13	68, 72, 79	4, 5, 6	6,000
12 UZD-1	81, 84, 83	7, 8, 9	55,000 (based on order for (2))
2 WTF-85	46, 51	10, 11	9,100
2 WTF-86	58	12	9,500
2 WTF-87	47, 51	19, 20	10,200
2 WTF-88	58	21	10,800
4 UNQ-11	71	13	8,800

Table 7.2.2 Worthington Centrifugal Pump Prices
(Refer to Table 3.2.1)

Pump Class	Efficiency %	Plant No.	Price
6 WTL-123	78	15	$30,000
6 UZD-1	68	15	17,500
4 WTF-124	72, 75, 76	14, 22, 23	21,000
6 WTF-124	78, 78, 78, 78 81	16, 17, 18, 24, 25	31,000
6 WTF-125	79, 82	26, 27	33,000
6 WTF-127	77	29	35,700
6 WTF-129	80	30	40,000
2-1/2 WTL-129	58	28	15,900

Table 7.2.2 Worthington Centrifugal Pump Prices (Continued)
(Refer to Table 3.2.1)

Equations representing the triplex plunger pump cost correlations are:

$$\$ = 219.5 \; (HP)^{.846} \qquad \text{Kobe}$$

$$\$ = 645 \; (HP)^{.363} \qquad \text{Gardner-Denver} \quad HP \leq 60$$

$$\$ = 5.51 \; (HP)^{1.522} \qquad \text{Gardner-Denver} \quad HP \geq 60$$

For plant size 10^5 GPD

These equations were not used in the computer programs. It was found to be simpler to enter discrete cost data and efficiency data for reciprocating pumps for the plant sizes for which these units may be used. Gardner-Denver costs were used for this purpose.

7.2.2 Centrifugal Pumps

The pricing of centrifugal pumps selected for the plant sizes under study became a more complex problem than was originally expected. The cost data that were available could not be readily transformed into equations relating dollars to an index characteristic of the pump.

The cost information is tabulated for each manufacturer in Tables 7.2.2 to 7.2.6 . These tables describe the type of unit, its efficiency for each plant application (the plant number code is given in Table 2.1.1), and the total pump cost including the following items:

- recommended material
- pump including lube oil supply console
- base-plate and coupling

Pump Class	No. Stage	Efficiency %	Plant No.	Price
1-1/2 VP	4	50, 53	1, 2	$ 5,000
1-1/2 VP	6	54, 51	3, 10	6,300
1-1/2 VP	7	53, 56	11, 12	7,000
1-1/2 VP	8	50, 53, 56	19, 20, 21	8,000
4 GTR	2	71, 73, 71	4, 5, 6	7,000
3 CNTA	8	72	13	14,000
5 HMTA	4 5 6	78 78 70	14 23 15	11,000 12,500 15,000
4 HMTA	7	74	22	13,000
3 HMTA	9	52	28	13,000
6 HMTA	5 7	78 78	24 (Lo P casing) 29 (Hi P casing)	13,500 18,000
8 HMTA	4	80, 79, 79	25, 26, 27	16,000
2-1/2 VHTB	17	65	28	25,000
14 GA	2	85 80	7, 8, 9, 16, 17, 18	100,000

Table 7.2.3 Ingersoll-Rand Pump Prices and Efficiencies

Pump Class	No. Stages	Eff. %	Plant No.	Price $
100 VLT	12	68, 70	1, 2	Not Available
	19	68, 70	10, 11	Not Available
200 VLT	14 22 30	70 70 70	3 12 12	Not Available
400 VLT	9	75	22	Not Available
500 VLT	11	80	4	Not Available
SD 6 x 8 x 10	6	63	3	Not Available
SMJ 4 x 6 x 15-1/2 HH	1	73	6	Not Available
DVDS 8 x 8 x 12	2	74	6	11,000
DVDS 14 x 16 x 18	2	75, 84.5, 85 82.5	18, 25, 26 27	25,000
DVSS 14 x 14 x 20	2	84	7	31,000
DVMX 4 x 6 9C	4	80, 78, 73, 80	13, 14, 15 22	4,400
DVMX 4 x 6 x 9D	4	79, 80	23, 24	4,400
DVMX 4 x 6 x 10B	5	78	24	5,600
DVMX 4 x 6 x 10C	7	81	29	8,400
DVMX 3 x 4 x 9A	12	57	28	8,400

Table 7.2.4 Byron-Jackson Pump Prices and Efficiencies
(These prices are for "standard" cast iron units)

Pump Class	No. Stages	Eff. %	Plant No.	Price $
5972	2	70, 74, 76.5	4, 5, 6	7,000

Table 7.2.5 Fairbanks-Morse Pump Price and Efficiencies

Part	PUMP CLASS										
	6 LB-15	6 LB-19	6 LB-22	12 LB-11	12 LB-12	10 LA-11	10 LA-12	15 LC-7	15 LC-11	VDM-12	VDM-14
First stage and barrel	6,400	10,695	10,695	12,760	12,760	8,990	9,090	18,195	18,195	9,940	9,940
Additional Stages	3,600	8,680	9,975	7,550	8,305	8,880	9,625	6,240	12,450	7,865	9,295
600 lb Flange	65	1,080	1,260	160	160	800	800	325	325	130	13[illegible]
Wearing Ring	570	351	408	759	828	616	665	693	1,052	1,008	1,176
Dynamic Balancing	180	—	—	187	205	—	—	154	242	—	—
Sleeve	75	—	—	104	104	—	—	126	126	90	90
Coupling	59	359	359	65	65	359	359	73	73	42	42
Seal	326	980	980	384	384	1,228	1,228	432	1,225	328	328
Total Cost $	11,275	22,145	23,677	21,969	22,811	20,873	21,767	26,238	32,688	19,443	21,001

Table 7.2.6 Peerless Pumps Pump Cost Calculations (Stainless Steel)

The material recommended for most of the selected pumps is 316 stainless steel. Because of the special material requirements for saline water application, the costs shown are 3 to 4 times higher than those for "standard" fabricated pumps.

When single-unit pumps were not readily available (e.g., for high pressure and very high flow rates), approximate single pump cost and efficiency values were assumed for use in the computer analysis.

The cost assumptions must be considered as rough approximations, since the number of units to be built is an important factor in setting the cost of a new pump design. If only one pump is developed and built to match new requirements, all of the costs of engineering development and design, patterns, and similar start-up costs must be recovered on that unit. If a larger market is expected, the start-up costs can be distributed over a number of units. These considerations are discussed in further detail in Section 10.

Reduction of the data from the manufacturers led to Table 7.2.7 which was used as the final cost input data for the computer programs. A number of different parameters were proposed in an effort to correlate the pump size to its cost. Most of the parameters used in these trials did not prove useful as correlating parameters. Some of the parameters considered were:

- specific speed
- parameter used in the hydraulic turbine cost correlation, $BHP/\sqrt{H}$
- the parameter described in Reference 14 and 15 as a "pump index" : $Q \times H^{1/2}$

The complexity of multi-stage centrifugal pump design, the difference in impeller diameters, and variations in efficiencies made these parameters inadequate for cost correlation. A final attempt led to a single cost correlation which related the dollar value of the pump to its discharge head for 1,000,000 GPD plants. The resulting correlation is shown in Figure 7.2.2. An average efficiency level was selected for each discharge pressure and capacity combination, and is presented in Table 7.2.7. It was

<table>
<tr><td colspan="4">10^6 GPD</td><td colspan="4">10^7 GPD</td><td></td></tr>
<tr><td>#4
$
68% - 6,000
71% - 7,000
70% - 7,000</td><td>#5
$
72% - 6,000
75% - 7,000
74% - 7,000</td><td>#6
$
79% - 6,000
71% - 7,000
75% - 7,000</td><td>W_x
I-R
F-M</td><td>#7
85 %
$ 100,000
1750 rpm →</td><td>#8
85 %
$ 100,000</td><td>#9
80 %
$ 100,000</td><td>I-R</td><td rowspan="2">400 PSIA</td></tr>
<tr><td>Average
70% - 7,000</td><td>Average
73% - 7,000</td><td>Average
75% - 7,000</td><td></td><td>Average
85% →
$ 100,000 →</td><td>Average</td><td>Average
80%</td><td></td></tr>
<tr><td>#13
$
71% - 8,800
72% - 14,000
80% - 13,200</td><td>#14
$
72% - 21,000
78% - 11,000
75% - 13,200
78% - 21,000</td><td>#15
$
78% - 30,000
68% - 17,500
70% - 15,000
73% - 11,000
78% - 30,000</td><td>W_x
I-R
B-J
P-P</td><td>#16
85 % →
$ 100,000 →
1750 rpm →</td><td>#17</td><td>#18
80%</td><td>R</td><td rowspan="2">600 PSIA</td></tr>
<tr><td>Average
74% - 12,000</td><td>Average
78% - 12,000</td><td>Average
71% - 12,000</td><td></td><td>Average
85 % →
$ 100,000 →</td><td>Average</td><td>Average
80%</td><td></td></tr>
<tr><td>#22
$
75% - 21,000
74% - 13,000
77% - 17,600</td><td>#23
$
76% - 21,000
78% - 12,500
79% - 17,600</td><td>#24
$
78% - 31,500
78% - 13,500
80% - 17,600</td><td>W_x
I-R
B-J</td><td colspan="3">#25 #26 #27
Average cost and efficiency deduced from parallel operation of units</td><td>I-R
W_x
B-J</td><td rowspan="2">800 PSIA</td></tr>
<tr><td>Average
75% - 18,000</td><td>Average
78% - 18,000</td><td>Average
78% - 18,000</td><td></td><td colspan="3">85 % - $ 300,000</td><td></td></tr>
<tr><td colspan="3">#29
77 % - $ 35,700
78 % - $ 18,000
81 % - $ 20,000</td><td>W_x
I-R
B-J</td><td colspan="3">Average cost and efficiency deduced from parallel operation of units</td><td></td><td rowspan="2">1500 PSIA</td></tr>
<tr><td colspan="3">78 % - $ 22,000</td><td></td><td colspan="3">85% - $ 300,000</td><td></td></tr>
</table>

Table 7.2.7 Data Used in the Computer Programs for Centrifugal Pump Cost Correlations

not possible to take into account all types of units recommended by the manufacturers because of the diversity of the data furnished.

The equation describing Figure 7.2.2 is:

$$\begin{cases} \$ = 2.02 \times (\text{Disch. Pressure, psia})^{1.36} \\ \text{Discharge Pressure, psia} \leq 800 \text{ psia} \end{cases}$$

and $\quad$ Plant Size = 10^6 GPD

For sea water operation at 1500 psia, the pump cost for the 10^6 GPD plant size was estimated to be $22,000 (Tables 7.2.2 to Table 7.2.7)

A uniform pump cost of $100,000 was selected from Tables 7.2.2 to 7.2.6 as representative of the cost of pumps used in plant sizes of 10^7 GPD with pump discharge pressures of 600 psia or less. For higher discharge pressure (800 and 1500 psia) at the 10^7 GPD plant size, the pump cost was estimated with reference to the cost of several pumps mounted in parallel which satisfy the plant conditions, or from estimates provided by manufacturers if one pump only was to be developed for a particular application. The cost estimate which was included in the computer program for these pumps was $300,000.

All of the cost values and average efficiencies used in the computer programs are shown in Table 7.2.7 for all pump speeds and types.

7.3 Hydraulic Turbines

Hydraulic turbines are not offered as standardized products by their manufacturers. Therefore, new units are designed for each application. Pricing information is derived from preliminary design work which is carried out for each new inquiry.

The James Leffel and Company firm provided pricing information for a number of Pelton units. Table 7.3.2 shows the design conditions for these units. The costs shown include the turbine and its required governor.

Table 7.3.3 shows the turbines offered by other manufacturers with the associated cost and efficiency data. The costs of governors have not been included in the turbine costs in this table. The costs shown for European manufacturers do not include import duties and freight costs.

The data in these two tables are summarized in Figure 7.3.1 which shows the cost correlations for hydraulic turbines. The cost curve was fitted by an equation which represents the capital cost of a Pelton Turbine without governors:

$$\$ = 10,300 \times \left(\frac{BHP}{\sqrt{H}} \right)^{.398}$$

This equation was used in the computer programs along with a turbine speed selection as follows:

Turbine Capacity Q (gpm)	Speed (rpm)
less than 200	3600
$200 \leq Q < 1500$	1800
$500 \leq Q < 3500$	1200
$3500 \leq Q$	900

Table 7.3.1 Pelton Turbine Speed Selection

This selection of hydraulic turbine speed was made in order to satisfy two conditions:

1. the specific speed of a Pelton turbine must be kept between 1.0 and 5.0 for good performance.

2. the cost of an electrical generator increases with both speed and power rating. Therefore, when the hydraulic turbine is used to drive a generator, the speed should be reduced as the power level increases.

Head (Ft.)	Flow (gpm)	EFF. %	R.P.M.	B.H.P. (HP)	N_S	$\frac{HP}{\sqrt{Head}}$	Turbine $	Gov. $
1765	6940	85.5	900	2650	4.05	63.1	53,000	11,500
3340	10500	87.0	900	7740	3.11	134	73,000	12,000
1590	1730	85.0	1200	590	2.9	14.8	30,000	8,500
1590	299	85.0	1200	102	1.21	2.55	15,000	5,700
1250	173	85.0	1800	46	1.64	1.3	12,000	2,000
1250	29.9	85.0	3600	8	1.76	.226	5,600	2,000
1675	29.9	85.0	3600	10.7	1.10	.261	6,000	2,000
822	173	85.0	1800	30.5	2.26	1.063	10,500	2,000
3175	104	85.0	3600	70.9	1.27	1.206	11,500	2,000
777	299	85.0	1800	49.0	3.07	1.81	13,000	2,000
732	1730	85.0	1200	270	5.19	10.0	26,000	5,700
1180	69.4	85.0	1800	176	3.45	5.12	20,000	5,700
1115	1730	85.0	1200	410	3.79	12.36	28,000	5,700
1675	2990	85.0	1200	1075	3.62	26.1	33,000	8,500
3000	1040	85.0	1800	670	2.1	12.25	29,000	5,700
2835	10400	85.0	900	6320	3.48	119.5	70,000	9,500

Table 7.3.2 Hydraulic Turbine and Governor Costs
For Various Head and Flow Specifications (James Leffel & Co.)

Manufacturer	Head (Ft.)	Flow (gpm)	Eff. %	RPM	Turbine $
Allis-Chalmers	3000	104	89.7	3600	16,000
	777	69.4	89.7	3600	16,000
	1675	173	89.7	3600	16,000
	867	2990	89.7	1200	35,000
	1675	2990	89.7	1200	35,000
Escher-Wyss G.M.B.H.	1115	694	83	3600	DM 42,000 ($10,500)
	1115	694	83	1800	DM 50,000 ($12,500)
	3175	10400	87.5	3600	DM 115,000 ($28,750)
	867	173	84	3600	DM 36,000 ($ 9,000)
	867	173	84	1800	DM 46,000 ($11,500)
	1675	2990	86	3600	DM 88,000 ($22,000)
	2835	104	80	3600	DM 43,000 ($10,750)
Voith GMBH	1765	694	87	3600	DM 60,000 ($15,000)
	1180	2990	85	1800	DM 78,000 ($19,500)
	1675	2990	88	1800	DM 84,000 ($21,000)

Table 7.3.3 Hydraulic Turbine Costs from Various Manufacturers

7.4 Driving Systems

7.4.1 Electric Motors

Cost data for electric motors are readily available from manufacturers' brochures. For squirrel-cage induction motors in the size range considered in this study, the chief manufacturers offer prices which are very close to one another for comparable motors. Listed motors have efficiencies of about 98%. The cost data presented below are taken from their price lists.

Squirrel-cage motors with horsepower ratings ranging from 200 HP to 20,000 HP have been considered. Data for three speeds: 400, 1800 and 3600 rpm, are given. At 1800 and 3600 rpm, the cost is directly proportional to the horsepower rating. At 400 rpm, the cost is proportional to the power below 3500 HP, and is an exponential function of the power above 3500 HP. In Table 7.4.1 the relations between these parameters and unit costs are shown. The total capital costs are shown in Figure 7.4.1 for motors in the range of interest.

Horsepower Range	RPM	Cost Relation
200 - 20,000	3600	$\$ = 14 \times HP$
200 - 20,000	1800	$\$ = 10.8 \times HP$
200 - 3,500	400	$\$ = 102.3 \times (HP)^{.809}$
3600 - 20,000	400	$\$ = 20.8 \times HP$

Table 7.4.1 Cost of Squirrel Cage Motors (Normal Starting Torque) (Allis-Chalmers, Peerless Electric)

7.4.2 Steam Turbines

Cost data for steam turbines are well documented in the sales information of the larger steam turbine manufacturers. For comparable types of equipment, the costs quoted by the different manufacturers are very similar. The cost data shown below were taken from data supplied by the Westinghouse Electric Corporation.

The costs of steam turbines are chiefly determined by their horsepower rating. . The most readily available turbines are those designed to operate at a speed of 3600 rpm. Slower speed models are not easily available and require the expense of custom designing. Higher speed models are readily available. Their costs are very similar to those at 3600 rpm for all speeds below 8000 rpm. Above this speed the cost increases with rpm.

Steam turbine prices are also affected by the steam conditions. For all turbines, the capital cost increases with increasing steam inlet temperature and pressures. For non-condensing turbines, the cost increases with increasing exhaust pressure. For condensing turbines, the capital cost decreases with increasing exhaust pressure.

For the present study, the steam conditions have been restricted to a narrow range of inlet and outlet conditions considered practical for reverse osmosis plant operation. These are summarized in Table 7.4.2.

Turbine Type	Inlet Conditions	Outlet Conditions
Non-Condensing	250 psi, 500 - 700° F	0 - 75 psig
Condensing	100 - 250 psi, 500° F	3 - 6 in. Hg. abs

Table 7.4.2 Steam Inlet and Outlet Conditions for Steam Turbines (Westinghouse Electric Corporation)

The costs of the two types of turbines are summarized in the form of equations in Tables 7.4.3 and 7.4.4. The condensing turbine costs include a cost of ten dollars per horsepower as the capital cost of the condenser unit. This number was obtained by verbal communication with the Westinghouse Electric Corporation. A plot of the total capital costs for the two types of turbines is shown in Figure 7.4.2.

The steam consumption of steam turbines is a function of both the steam thermodynamic conditions and the turbine efficiency. The heat available for use by

Horsepower Range	Cost Base Cost + Variable Cost + Condenser Cost
less than 500	19,500
500 - 1,000	19,500 + 11.4 (HP - 500) + 10 x HP
1,000 - 1,500	20,500 + 11.4 (HP - 500) + 10 x HP
1,500 - 2,000	21,500 + 11.4 (HP - 500) + 10 x HP
2,000 - 2,500	40,100 + 17.7 (HP - 2000) + 10 x HP
2,500 - 3,000	42,100 + 17.7 (HP - 2000) + 10 x HP
3,000 - 3,500	62,300 + 24.1 (HP - 3000) + 10 x HP
3,500 - 4,000	65,300 + 24.1 (HP - 3000) + 10 x HP
4,000 - 10,000	46,500 + 22.1 (HP - 1000) + 10 x HP
10,000 - 20,000	243,200 + 12.9 (HP - 10,000) + 10 x HP

Table 7.4.3 Condensing Steam Turbine Capital Costs
(Westinghouse Electric Corporation)

Horsepower Range	Cost Base Cost + Variable Cost
less than 500	17,500
500 - 1,000	17,500 + 9.4 (HP - 500)
1,000 - 1,500	18,500 + 9.4 (HP - 500)
1,500 - 2,000	19,500 + 9.4 (HP - 500)
2,000 - 2,500	35,100 + 16.8 (HP - 2000)
2,500 - 3,000	37,100 + 16.8 (HP - 2000)
3,000 - 3,500	56,400 + 24.1 (HP - 3000)
3,500 - 4,000	59,400 + 24.1 (HP - 3000)
4,000 - 10,000	42,500 + 22.3 (HP - 1000)
10,000 - 20,000	222,650 + 11.75 (HP - 10,000)

Table 7.4.4 Non-Condensing Steam Turbine Capital Costs
(Westinghouse Electric Corporation)

turbines is shown in Table 7.4.5 for 500° F steam at three inlet pressure levels. For the condensing turbine, the outlet condition is 3 inch Hg (absolute) and 115° F. For the non-condensing turbine, the steam exhausts near saturation at 50 psia.

Steam Inlet Pressure	Heat available, Btu/lb steam Condensing	Non-Condensing
100 psig	308.7	39.2
150 psig	329.6	73.6
250 psig	353.7	108.3

Table 7.4.5 Heat Available from 500° F Steam

The consumption of steam during turbine operation is:

$$\frac{lb_{steam}}{HP - hr} = \frac{2544}{\text{heat available x turbine efficiency}}$$

Turbine efficiencies vary between 60% to 90%. For 100 - 300 psig inlet steam, the efficiency can be calculated by this equation:

$$\log(\text{efficiency}) = .099 \log(HP) - 1.166$$

The equation applied, within 5 points, to both condensing and non-condensing turbines.

7.4.3 Diesel Engines

Diesel engines available in the standard lines of products from the major manufacturers are similar in capital costs. These engines cost approximately $82 per horsepower for engines smaller than 2500 HP, and cost $93 per horsepower for engines between 2500 and 20,000 horsepower.

Maintenance costs of diesel engines are approximately $1 per horsepower-year. Fuel consumption is approximately 0.37 pounds of fuel per horsepower - hour. Lubricating oil consumption is 1 - 2 gal. per horsepower - year.

Diesel Engine Size (HP)	Cost
HP < 2500	$ = 82 x (HP)
800 ≤ HP ≤ 20,000	$ = 93 x (HP)

Table 7.4.6 Diesel Engine Cost Correlations
(Fairbanks - Morse)

7.5 Electric Generators

Cost data for electric generators were provided by the Westinghouse Corporation and Power Equipment Company. The costs differ slightly depending on the ratings of the machines selected. Table 7.5.1 shows the resulting cost correlations broken down according to generator power, speed, and voltage ratings. Figure 7.5.1 is a graphical representation of the equations derived from sales catalog data.

7.6 Gears

Gear costs have not previously been well documented in terms of their cost versus power, input rpm, and speed ratio. When buying gears for large machinery, the gears are generally designed and tailored to meet the requirements within specified tolerances. For this reason, gear costs have not previously been generalized.

Data for gear costs versus rpm and speed ratio at two power levels for gears thought to be "typical" of the type to be used in a desalination plant were generated especially for this project by the Western Gear Corporation. These data are shown in Figures 7.6.1 and 7.6.2. Based on these data, an equation for gear cost was developed as follows:

$$\text{Gear cost} = 5.23 \left[HP\right]^{0.903} \left[1 + 1.562\ R\right] \left[\frac{400}{RPM}\right]^{.5}$$

where:

RPM = revolutions per minute of the smaller gear

360 rpm	
8000 → 900 KW	$\$ = 140.06\ (KW)^{.812}$
900 → 200 KW	$\$ = 546.45\ (KW)^{.612}$
400 rpm	
8000 → 600 KW	$\$ = 140.06\ (KW)^{.810}$
600 → 200 KW	$\$ = 568.18\ (KW)^{.590}$
600 rpm	
8000 → 400 KW	$\$ = 230.41\ (KW)^{.840}$
400 → 125 KW	$\$ = 440.53\ (KW)^{.586}$
1800 rpm	
1000 → 600 KW	$\$ = 35.71\ (KW)^{.986}$
600 → 350 KW	$\$ = 19.76\ (KW)^{.855}$
1800 rpm	
30 → 350 KW	$\$ = 272.48\ (KW)^{.592}$
900 rpm	
8000 → 500 KW	$\$ = 84.75\ (KW)^{.841}$
500 → 175 KW	$\$ = 124.38\ (KW)^{.781}$
1200 rpm	
8000 → 5000 KW	$\$ = 33.11\ (KW)^{.936}$
5000 → 350 KW	$\$ = 93.28\ (KW)^{.815}$

Table 7.5.1 Electric Generator Cost Correlations

R = gear speed ratio (speed of smaller/speed of larger)

HP = power input

Cost = initial capital cost

Gear efficiencies are generally about 98%.

7.7 Plant Costs

Typical costs for those aspects of desalination plant operation which are of interest for the present investigation were provided by the Office of Saline Water (OSW Report No. 134). These costs are:

- Membrane capital cost (exclusive of the pressure vessel):
  $ .25/ft^2 (brackish or sea water)

- Labor cost:
  $2.50/hour

Capital costs were assumed to be amortized over a period of 30 years as follows:

5.27% based on 3-1/4 % interest for 30 years

plus .25% charge for insurance

7.8 Service and Maintenance Costs

7.8.1 Pumping System

The "operating cost" for the pump is composed of two different items:

1. Spare Parts - The cost of spare parts that would be needed during 30 years of operation were estimated by the manufacturers to be roughly 2/3 of the original capital cost of the unit. This cost was treated as a capital cost and, as such, was amortized over 30 years.

2. Service Cost - The cost of servicing the pump was estimated to be \$ .25/operating hour based upon the following labor estimates. In order to maintain and service the pump, which is the key element of the pumping system, it was assumed that two men would be required 8 hours/day and 50 days/year. This time allows for inspection periods, overhauling the pump during plant shutdowns and any breakdown periods, and for replacement or repair of defective parts.

This service cost estimate agreed with one figure provided by a manufacturer. This information, however, is only a rough estimate. No manufacturer could provide us with actual service or maintenance cost data.

The spare parts described in Item (1) above will include a spare rotor to minimize the length of time the pump will be out of service if the rotating parts need repairs. The defective parts can be replaced at the factory while the unit continues in service using the spare rotor.

7.8.2 Driving System

The service and maintenance costs for an electric motor drive were assumed to be negligible.

Service and maintenance costs for steam turbines were calculated from manufacturers information. The resulting figure is \$ .10/operating hour.

Service and maintenance costs for diesel engines were based upon data gathered from manufacturers. The total service and maintenance costs amount to:

\$2.30/HP - year

including \$1.0/HP - year (for maintenance)

2 gallons/Hp - year of lube oil
at \$ .65/gallon

7.8.3 Membranes

The cost provided by the Office of Saline Water for the replacement of the membranes was:

$ .50/ft^2 - during the membrane operating lifetime

A one-year operating lifetime may be assumed for the membranes used in brackish water desalination plants. A six-month operating lifetime may be assumed for membranes to be used in sea water desalination plants.

7.8.4 Energy Recovery Systems

The operating cost for the energy recovery system is composed of two different items:

1. Spare Parts: The costs of spare parts required for the hydraulic turbine during an operating period of 30 years were roughly estimated to be 2/3 of the original capital cost of the unit. This cost was treated as a capital cost and , as such, was amortized over 30 years.

2. Service and Maintenance Costs: These costs for hydraulic turbines were based on assumptions similar to those made for pumps in Section 7.8.1. However, since the hydraulic turbine operation is less critical than the operation of the pump to the continued production of water, the two men were assumed to be required to service the turbine only 20 days/year. This service cost amounts to:

   $ .10/operating hour

7.9 Power Costs

The figures shown below were provided by the Office of Saline Water (Report No. 134) and, for the fuel cost, by a diesel engine manufacturer:

- electricity: $ .007/KW-hr (OSW)

–	steam:	$ .55/10^6 BTU	(OSW)
–	steam resale price:	$ .15/10^6 BTU	(manufacturer's estimate)
–	diesel fuel:	$ .06/gallon (= $ .01/lb)	(manufacturer's estimate)

7.10 Power Consumption Calculations

The cost of the power required to drive the pumping system is based upon the horsepower (HP) input to the pump.

– Electric Motor Drive:

The cost of electricity was given as 0.7 ¢/KW-hr in the preceding section. The calculation of KW-hr/1000 gallons of fresh water output is accomplished by the following equation:

$$\frac{\text{KW-hr}}{\text{1000 gallons}} = \frac{\text{(HP input)}}{\text{(El. Motor Eff.)}} \times (.7457) \times \frac{24\ \text{(hr)} \times 1000}{M_{F.W.}\ \text{(GPD)}}$$

– Steam Turbine Drive:

The cost of steam was given as 55¢/million BTU in the preceding section. The steam turbine is sized accordingly to the horsepower input to the pump and the turbine efficiency. The calculation of BTU/1000 gallons of fresh water output that are consumed is carried out using the following equations:

$$\text{(St. Turb. Eff.)} = \text{(HP input)}^{.099}/3.2$$

$$\frac{\text{BTU}}{\text{1000 gallons}} = \frac{\text{(HP input)}}{\text{(St. Turb. Eff.)}} \times (2545) \times \frac{24\ \text{(hr)} \times 1000}{M_{F.W.}\ \text{(GPD)}}$$

– Diesel Engine Drive:

The diesel output horsepower must be equal to the pump input horsepower. The cost of fuel was given as 1¢/lb in the preceding section. The diesel fuel consumption rate was established at

0.37 lb/hp-hr from manufacturers' information. The calculation of lb/1000 gallons of fresh water output is represented by the following equation:

$$\frac{\text{lb}}{1000 \text{ gallons}} = .37 \times (\text{HP input}) \times \frac{24 \text{ (hr)} \times 1000}{M_{F.W.} \text{ (GPD)}}$$

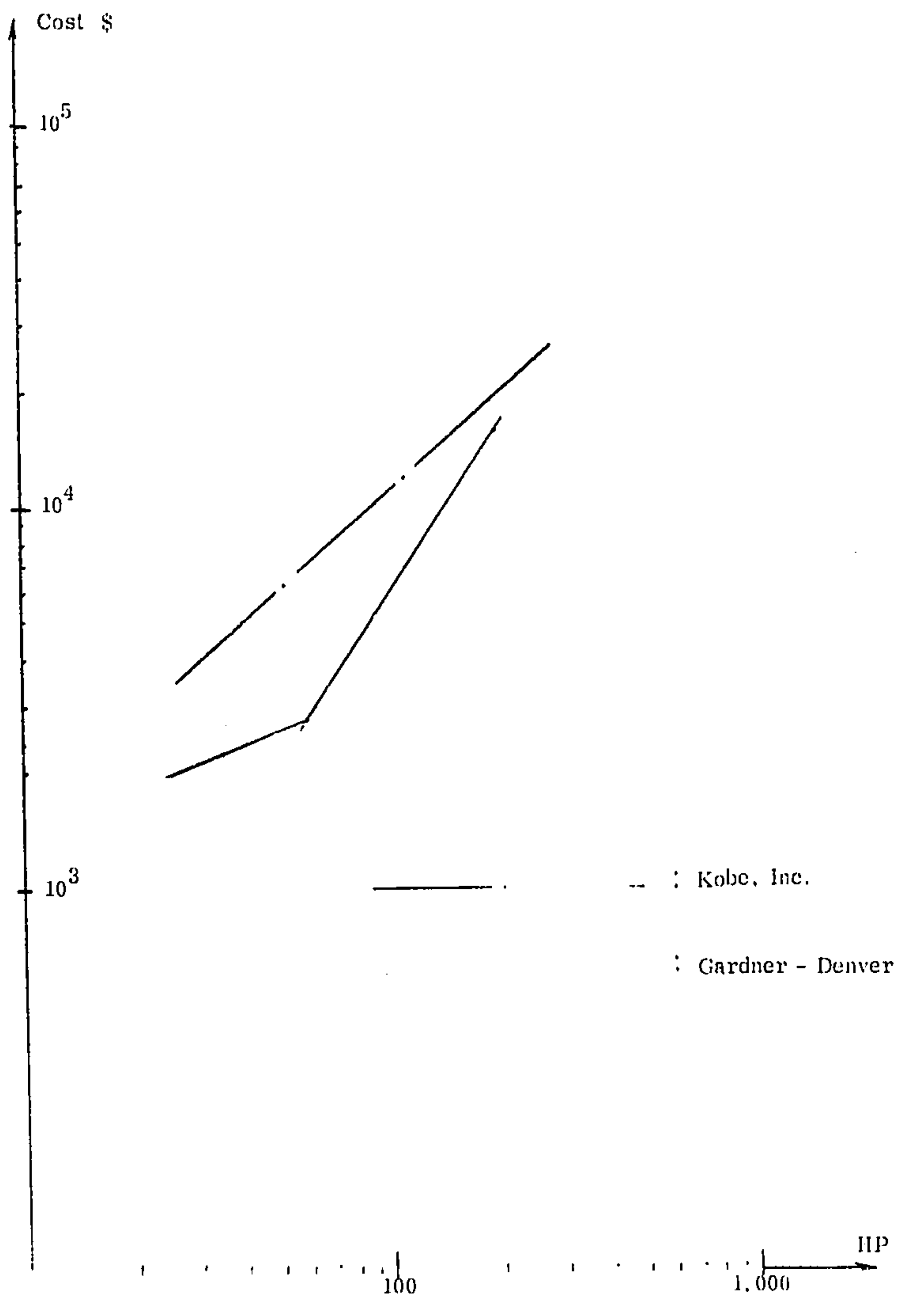


Figure 7.2.1 Reciprocating Pump Cost Correlations

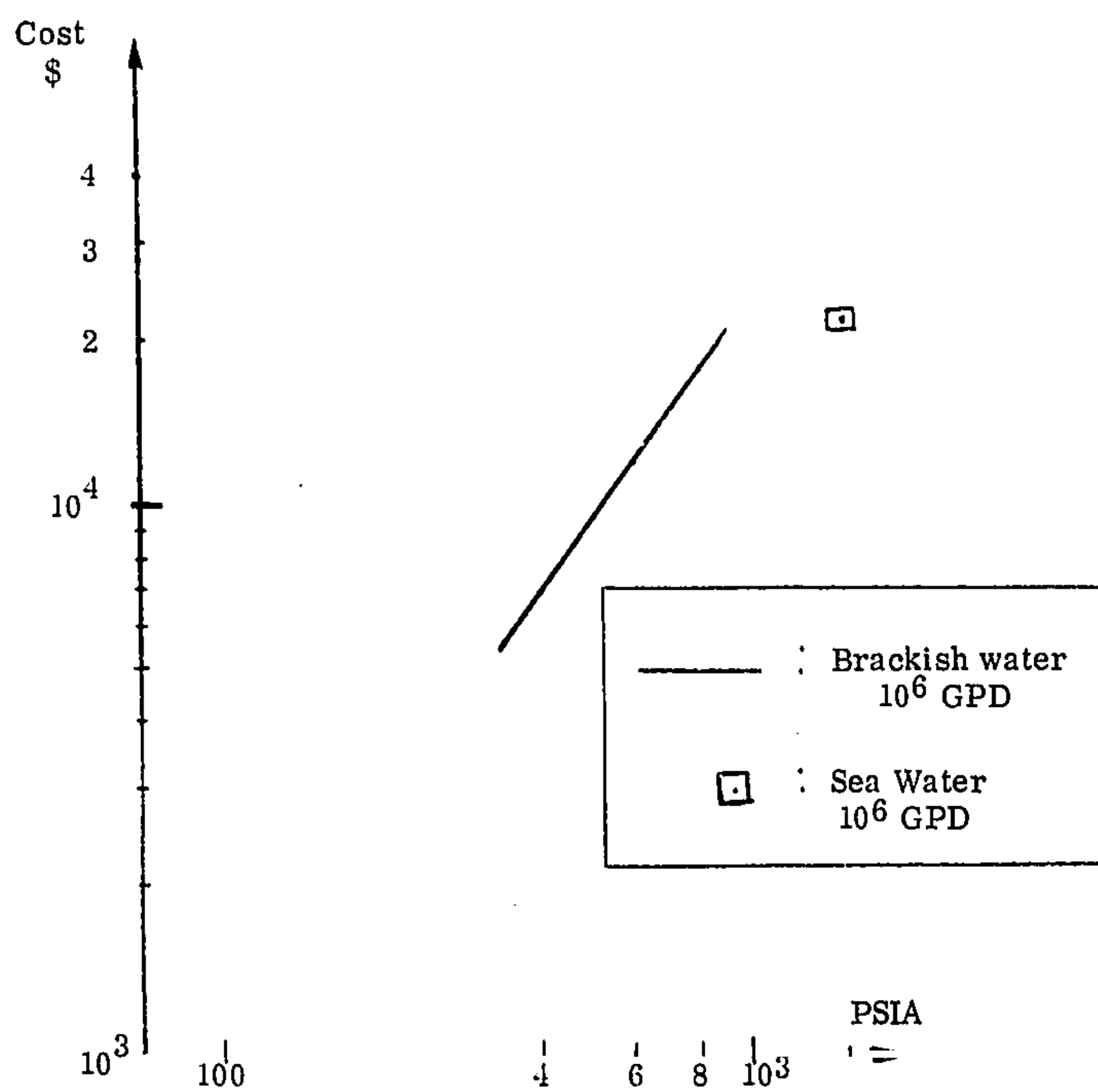


Figure 7.2.2 Centrifugal Pump Cost Correlations
Used in Computer Programs

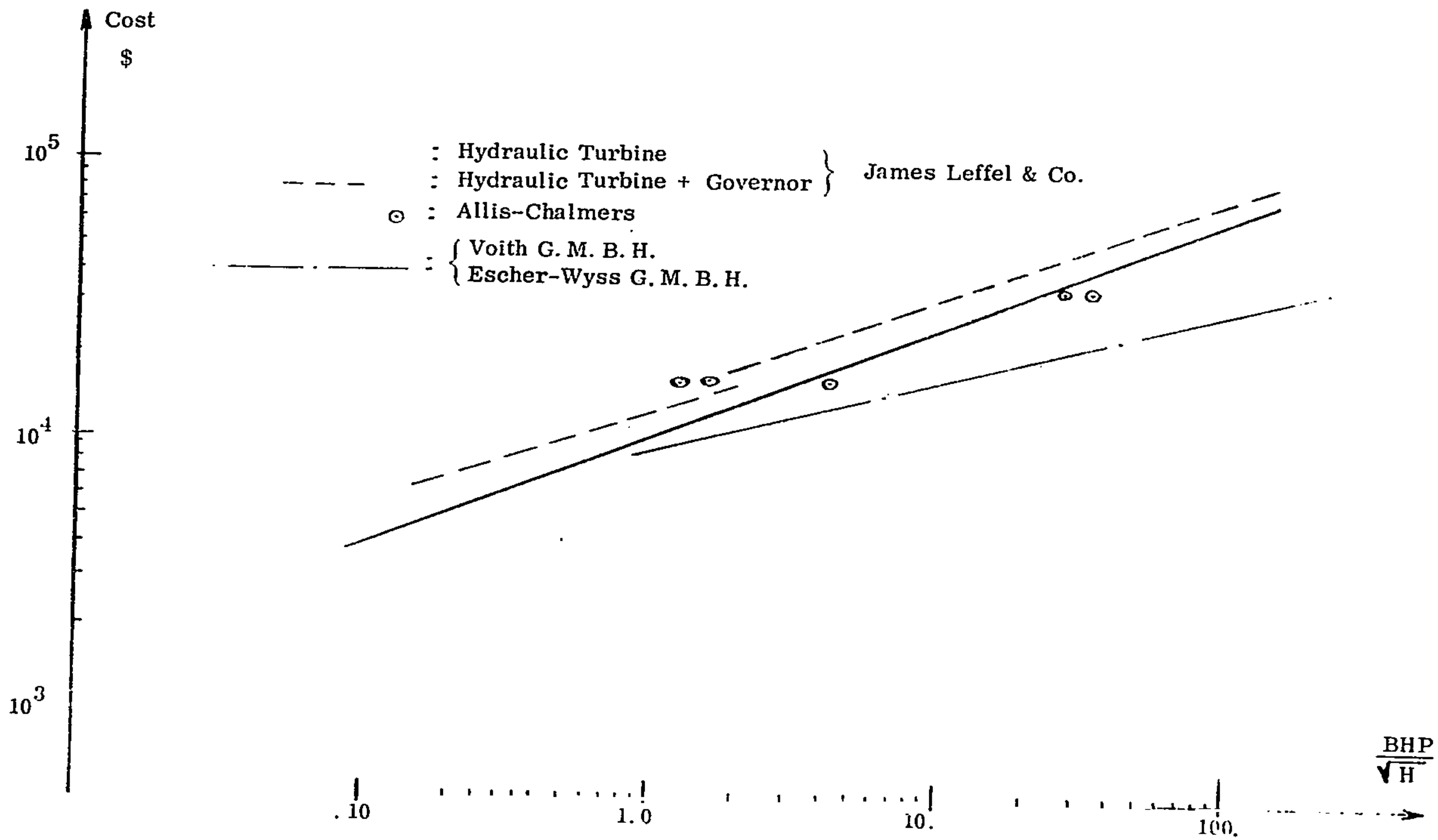


Figure 7. 3. 1 Hydraulic Turbine Cost Correlations

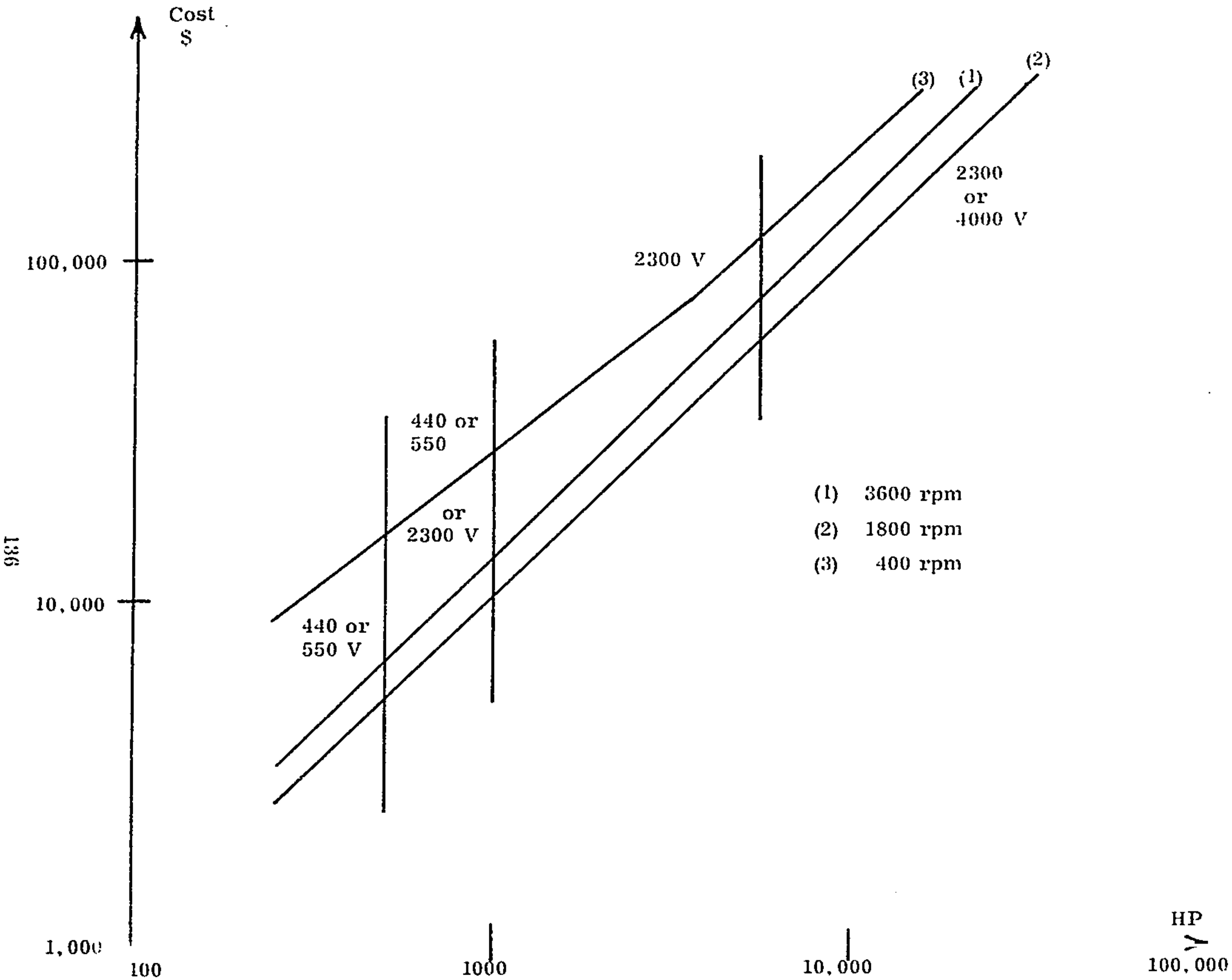


Figure 7.4.1 Normal Starting Torque Induction Motor Cost Correlations

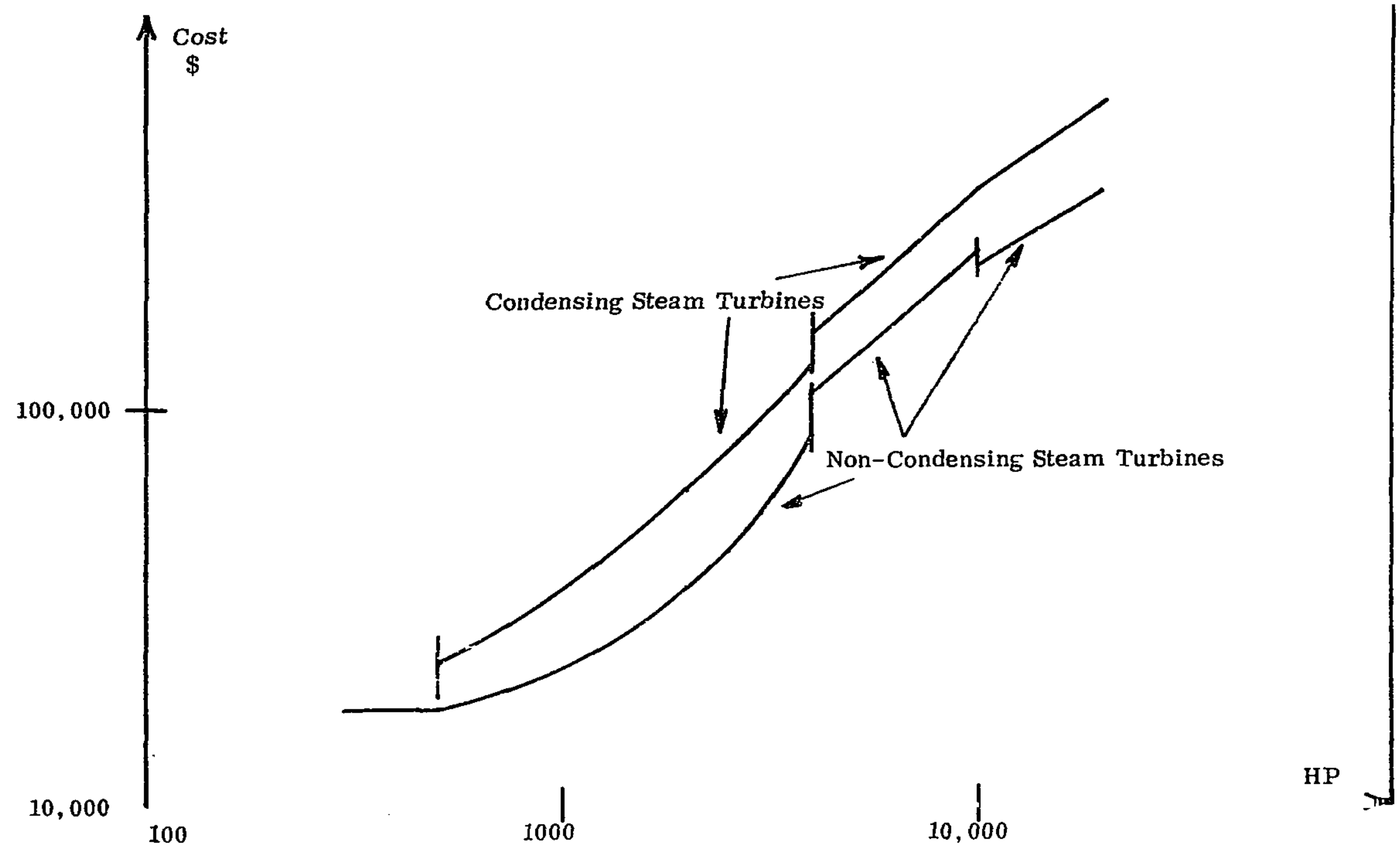


Figure 7.4.2 Steam-Turbine Cost Correlations
(From Tables 7.4.3 and 7.4.4)

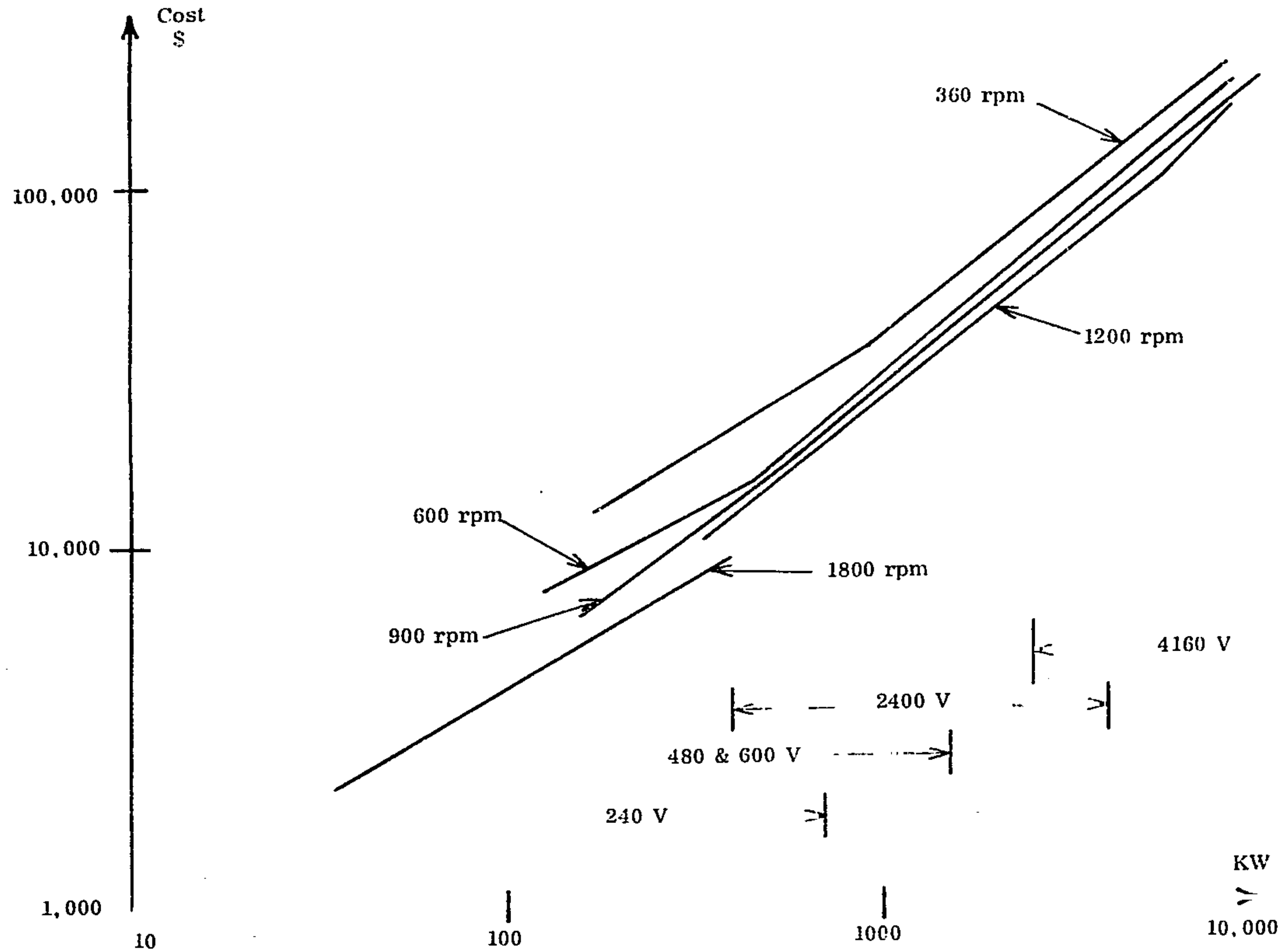


Figure 7.5.1 Electrical Generator Cost Correlations
(From the Westinghouse Electric Corp., and Power Equipment Co.)

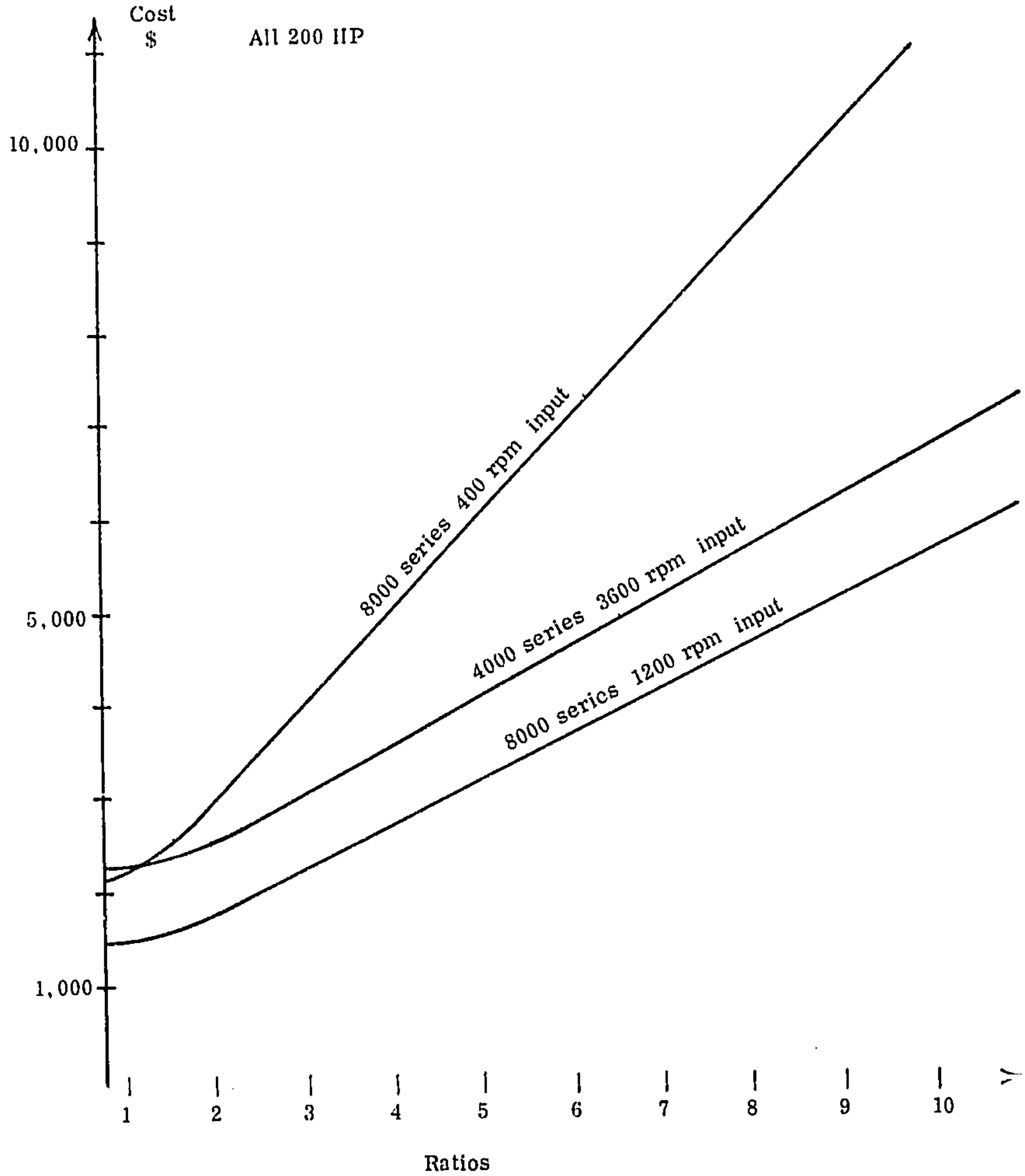


Figure 7.6.1 Gear Cost Correlations
(From Western Gear Corporation)

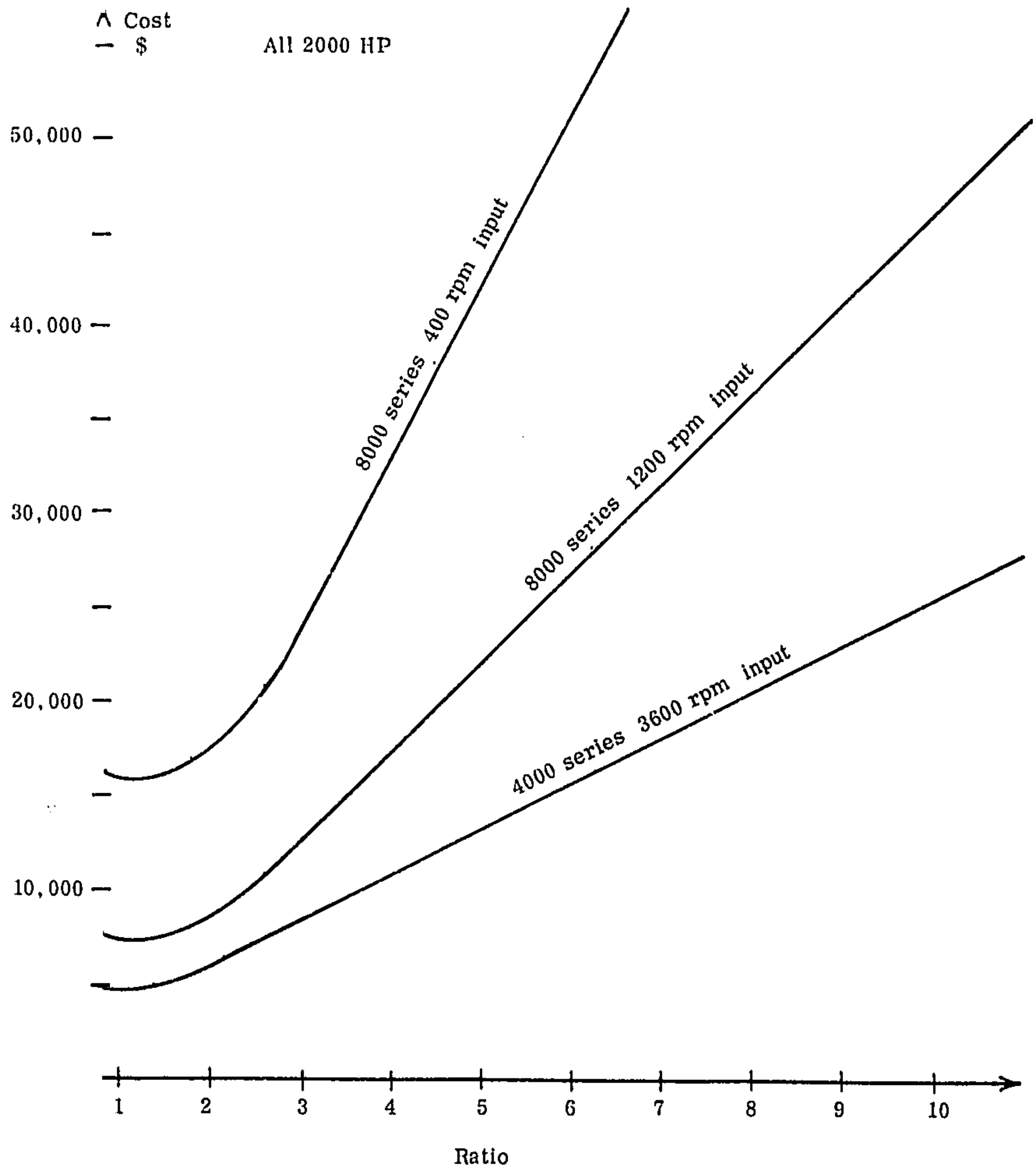


Figure 7.6.2 Gear Cost Correlations
(From Western Gear Corporation)

Section 8

COST ANALYSIS COMPUTER PROGRAMS

8.1 Introduction

In order to compare the various plant arrangements for the selected "standard" plants and to select optimum configurations, two computer programs have been written. These programs are written in Fortran IV language for use on the CDC 3600 computer.

Four drivers and two types of power recovery systems have been considered. Electric motors, non-condensing and condensing steam turbines, and diesel engines have been included in the study. The first power recovery system utilizes a hydraulic turbine driven by the concentrated brine leaving the membranes and is connected to the pump through a gear reduction unit. A sketch of this arrangement is shown in Figure 8.1.1 (a). When comparing this system with power recovery to a plant without power recovery, two different drivers must be considered. A smaller driver is required for the pump when power recovery is utilized.

In the second type of power recovery system, the hydraulic turbine drives an electric generator through a gear box (Figure 8.1.1 (b)). The first computer program for this power recovery arrangement considers only electric motors as drivers. The electricity generated is fed back into the motor. The motor size is unaffected by the addition of the energy recovery system but a smaller amount of power will be required from outside sources. This program performs a speed matching optimization. Six generator speeds and three electric motor speeds are considered and the program calculates all possible combinations and their corresponding costs.

The second program contains three subroutines to compute various costs and operating requirements associated with the two types of steam turbine drives and the diesel engine drive. Again, when using the first type of power recovery system (direct gearing), two sizes of drivers are considered. With the second type of power recovery system, the electricity produced is assumed to be sold outside the plant.

Capital costs for the various pieces of equipment have been obtained from manufacturers' data as discussed in Section 7. The data was reviewed and equations expressing the cost as a function of some parameter (e.g., horsepower) have been formulated, or else costs for various size, speed, head, or capacity ranges have been established. These costs were amortized over a thirty-year period and insurance costs have been included.

Typical operating costs and power costs have been estimated, as described in Section 7, to allow realistic evaluation of the economic advantages of power recovery in large size desalination plants.

Separate sets of programs have been written for brackish water input and for sea water input. The major differences between "sea water" programs and "brackish water" programs are the plant design requirements listed in detail in Section 2.1. The subprogram that describes pump costs has also been changed, reflecting the increased cost and operating efficiency of the sea water units. The membrane capital and operating costs also are changed since a six-month operating life was assumed for sea-water reverse osmosis membranes, while brackish water membranes are expected to operate for a year. However, the same basic program structure was retained for both sets of programs.

8.2 Typical Programs

8.2.1 Main Programs and Subroutines

To illustrate the operation of the computer programs in greater detail, a flow chart (Figure 8.2.1) and a listing of the Fortran deck is given below. The program shown is for plants 2, 3, and 5. It contains subroutines for the non-condensing and condensing steam turbines and the diesel engines as drivers.

Two code variables have been used in the program:

PLTC = plant power recovery system variable

PLTC = 0.0 if the power recovery system with electric generator is being considered (Figure 8.1.1 (b))

PLTC = 1.0 if the power recovery system with direct gearing is being considered (Figure 8.1.1 (a))

2) DC = driver code variable

DC = 2.0 for non-condensing steam turbines (Plant 2)

DC = 3.0 for condensing steam turbines (Plant 3)

DC = 5.0 for diesel engine driver (Plant 5)

The dummy variable DC = 0.0 is placed at the end of the driver code (DC) data cards to insure a logical stop for the program.

The program requires only the driver code to be read in as data. If additional drivers are to be studied in the future, new subroutines and call statements would be necessary. The program can be modified to permit analysis for different plant specifications by changing the cards in the appropriate section of the program.

The electric motor program is similar except that the electric motor cost calculations are included as part of the main program and not as a subroutine. A large number of comment cards have been used throughout the program to make the listing more understandable.

8.2.2 Description of Program Outputs

The output format for the two programs (Figure 8.2.2 and Figure 8.2.3) differ in certain respects. The plant specifications (GPD, feed pressure, recovery fraction, and fractional pressure loss), membrane data, and pump and hydraulic turbine information are identical for both plants. The cents/1000 gallons cost given is the contribution of the capital cost to the total water cost. (Capital cost input data has been described in Sections 7.2, 7.3, and 7.7).

In the program for plants 2, 3, and 5, the driver information is given next. (The input data has been described in Sections 7.4.2, and 7.4.3). In cases where two lines of data are given, the second is for the case where a direct gear between the pump and the power recovery system is being considered. The total cost given includes both capital and operating costs.

The output data listed below the driver information is divided into four headings. Several columns are included under the last three headings.

- SPEED, RPM - lists the generator speeds considered in the study.

- OPERATING COSTS - lists the total operating costs in dollars for the pump and the hydraulic turbine. (Sections 7.8.1 and 7.8.3).

- CAPITAL COSTS - lists the capital costs in dollars for the gear sets and the generator. GEARS 1 gives the total cost of all gear sets in the plant when power recovery is used. GEARS 2 gives the total cost in the plant without power recovery. The operating costs for the gears and generator were assumed to be negligible. (Sections 7.5 and 7.6).

- TOTAL CONTRIBUTIONS TO WATER COST - lists the total cost including capital and operating costs (where applicable) on a cents/1000 gallon basis for the pump, hydraulic turbine, gears, and generator. TOT1 is the total plant cost (including power costs) for the plant when power recovery is used; TOT2 is for the plant without power recovery. DELTA is TOT2 - TOT1 and, if positive, indicates that power recovery will be profitable.

The power cost is listed for both plants with and plants without power recovery. In the case where the power recovery system has a generator, the electricity resale value has been deducted from the cost of power supplied to the driver. (Section 7.9).

In the case where the hydraulic turbine is directly coupled to the pump, the generator speed, capital cost, and contribution to water cost are zero. The power cost in either case is the cost of the power supplied directly to the driver.

In the electric plant program, the information listed after the hydraulic turbine data is broken down as follows:

- SPEEDS, RPM - lists the speeds of the motors and generators which were studied.

- OPERATING AND CAPITAL COSTS - The first two columns give the operating costs in dollars for the pump and hydraulic turbine. The next four columns give the capital costs in dollars for the gear sets, the electric motor (Section 7.4.1) and the generator. GEAR 1 couples the motor and the pump while GEAR2 couples the hydraulic turbine and the generator. Operating costs for these four items were assumed to be negligible.

- CONTRIBUTIONS TO WATER COST - The first two columns list the operating costs for the pump and the hydraulic turbine. To find the total contribution of either item, add this cost to the capital cost listed above. The next two columns give total costs for the motor and generator since operating costs were assumed to be zero for these machines. TOT1 is the total plant cost (including power) when power recovery is used and TOT2 is the total cost without power recovery. DELTA is TOT2 - TOT1 and, if positive, a power recovery system will be a profitable addition to the plant.

In the case where the hydraulic turbine is coupled to the pump, the generator speeds and costwill be zero, and the motor costs will be those of a smaller motor. In computing TOT2, the cost of a larger motor is used.

In the plant where the power recovery system produces electricity, Program 4, the power cost with recovery reflects the reduced amount of electrical power that must be purchased outside, while in Program 1 the reduced power cost with recovery is due to the use of a smaller electric motor.

8.2.3 Speed Matching

Many speed combinations are possible for the driver, pump, power recovery turbine, and (when included) the electrical generator used in a given pumping system arrangement. An iteration procedure has been included in each computer program to explore each of the possible speed combinations that can be used to satisfy the plant design requirements. When necessary, the computer takes into account the need for gears or belts with their associated costs. All possible combinations are calculated and printed out. The optimum combination, or combinations, can be selected by scanning the column that describes the total contribution of the pumping system to the water cost.

The speeds used in the program have been chosen as the most typical speeds for each type of machine from the information obtained from the manufacturers. If, for example, consideration of a different pump speed appears to be warranted as a result of future developments, the new speed value can be added to the program and the computer will analyze the resulting new combinations.

Figures 8.2.2 and 8.2.3 are typical outputs illustrating the speed matching in cases 1 and 4.

8.3 Program Listing

One typical program listing is given in the next pages. This listing represents the optimization program used for arrangements A and B as described in Section 6.2.1. These arrangements are electric motor driven sea water plants with and without energy recovery.

```
*JOB,FINK,21952,,03,4444
*EQUIP,6=61
*FTN,L,X
      PROGRAM PLANT4SH
      DIMENSION PROD(3),P(3),R(3),PL(4),SP(3),SM(10),ST(10),SP(10),
     1SG(10)
      CGR(HP,GR,S)=5.23*(HP**.933)*(1.+1.552*GR)*((400./S)**0.5)
  100 FORMAT(1H1)
  101  FORMAT(1H1,4HGPD=F9.0,5X,4HPSI=F4.0,5X,9HRECOVERY=F3.1,5X,16HFRAC
     1TION P-LOSS=F4.2//)
  106  FORMAT(1X,7HHEAD,FT,4X,13HFLOW RATE,GPM,4X,9HSPEED,RPM,4X,10HEFFI
     1CIENCY,4X,10HHORSEPOWER,4X,14HCAPITAL COST,$,4X,13HCENT/1000 GAL)
  107  FORMAT(1X,F7.1,6X,F7.1,9X,F6.0,8X,F5.2,8X,F7.1,9X,F7.0,11X,
     1F5.1//)
  102 FORMAT(120H  SPEEDS,RPM  **  OPERATING   AND   CAPITAL COSTS,DOLLA
     1RS      ** CONTRIBUTIONS TO WATER COST, CENTS/1000GAL           )
  108  FORMAT(1X,28HMEMBRANE     THRUPUT,GPD/FT2=F3.0,4X,14HCAPITAL COST=
     1$F8.0,4X,14HCENT/1000 GAL=F5.1//)
  109  FORMAT(//,1X,21HPOWER (CENT/1000 GAL),4X,11HWITH RECOV=F5.1,4X,14
     1HWITHOUT RECOV=,F5.1)
  110  FORMAT(1X,9HPUMP DATA)
  111  FORMAT(1X,12HTURBINE DATA)
  104 FORMAT(120H     MOT    GEN     PUMP     TURB     GEAR1    GEAR2    MOTOR
     1    GEN         PUMP  TURB  MOTR   GEN  TOT1   TOT2  DELTA
  103 FORMAT(2X,2F6.0,6F8.0,4X,7F6.1)


C                 PLANT 4 -  ELECTRIC POWERED WITH GENERATOR
C           PLANT SPECIFICATIONS
C      1-   PLANT CAPACITIES , GPD
       PROD(1)=100000.0
       PROD(2)=1000000.0
       PROD(3)=10000000.
C      2-   FEED PRESSURES, PSIA
       P(1)=1500.
C      3-   FRACTIONAL RECOVERY , PRODUCT/FEED
       R(1)=0.4
C      4-   FRACTIONAL PRESSURE LOSS ALONG MEMBRANE BASED ON FEED PRESS.
       PL(1)=0.
       PL(2)=0.05
       PL(3)=0.10
       PL(4)=0.15


C           SPECIFIED UNIT COSTS
C      AMORTIZATION RATE -- FRACTION OF CAPITAL COST
       AMT=0.0527
C      INSURANCE RATE -- FRACTION OF CAPITAL COST
       CINS=0.0025
C      SPECIFIED UNIT COST
C      ELECTRICITY COST , DOLLARS / KWH
       CELE=0.007
C      MEMBRANE COST,FIXED PRICE OF STRUCTURE,DOLLARS/SQUARE FOOT
       CMEM=0.25
C      MEMBRANE OPERATING COST,DOLLARS/SQUARE FOOT-YEAR
C      MEMBRANE HAS 6 MONTH OPERATING LIFE IN SEA WATER
       CMOP=0.5
C      THROUGHPUT RATE IN GPD/SQUARE FOOT OF MEMBRANE
       SP(1)=10.

C           PERFORMANCE ESTIMATION
       PLTC=0.4
  501  J=1
```

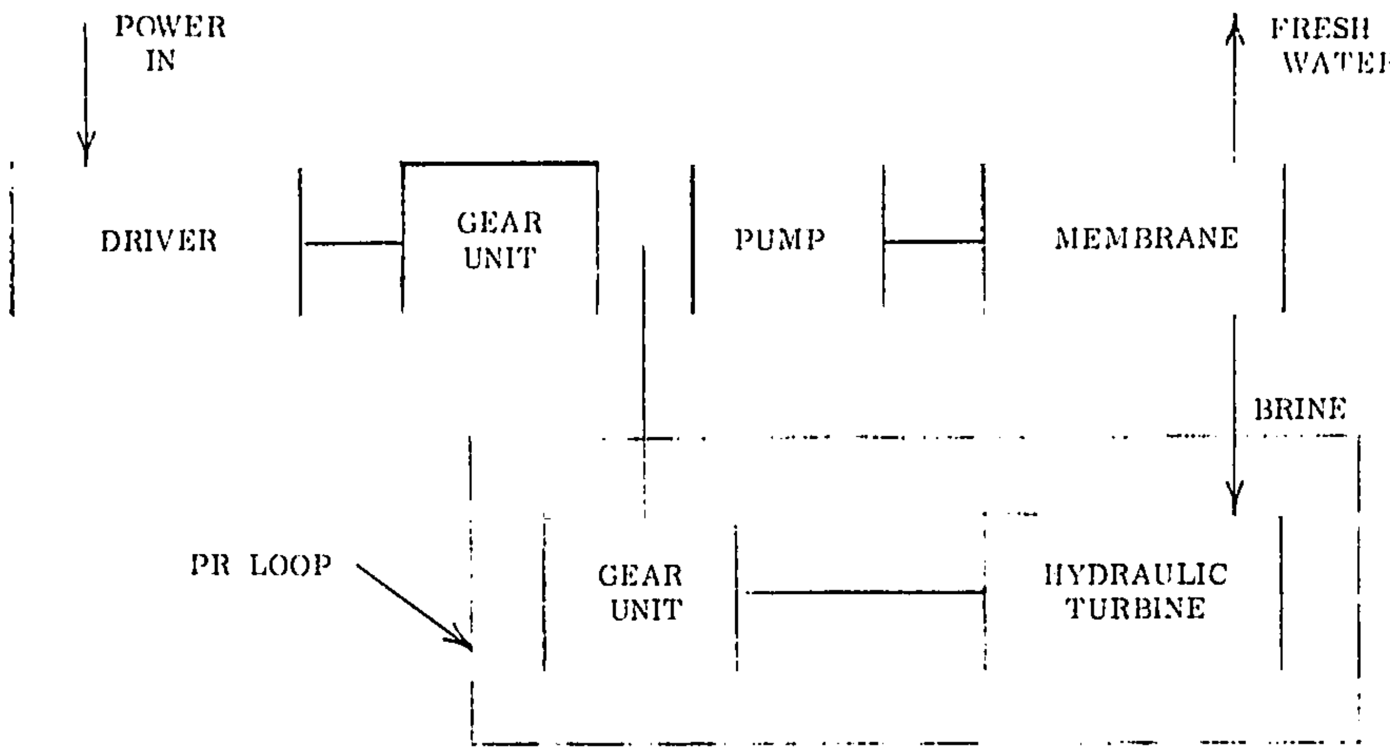
```
      K=1
      DO 6 [illegible] =1,3
      DO 6 [illegible] L=1,4
C     CALCULATION OF REQUIRED FEED RATE , [illegible]
      FR=PROD(I)/R(K)
C     CALCULATION OF PUMP OUTPUT POWER , HP
      HP=[illegible]5*(P(J)-15.)*FR/100[illegible].
C     FLOW RATE THROUGH PUMP,GPM
      GPMP=FR/24./60.
C     CALCULATION OF PRESSURE HEAD ON PUMP , FT
      HPP=(P(J)-15.)*144./64.
C     BRINE FLOW RATE THROUGH HYDRAULIC TURBINE,GPD
      B=(1.-R(K))*FR
C     FLOW RATE THROUGH TURBINE, GPM
      GPMT=B/24./60.
C     PRESSURE OF BRINE FEED INTO TURBINE,PSIG
      PHT=(1.-PL(L))*(P(J)-15.)
C     CALCULATION OF PRESSURE HEAD OF [illegible] INTO TURBINE, FEET
      HPHT=PHT*144./64.
C     CALCULATION OF HYDRAULIC TURBINE INPUT HORSEPOWER
      HPHT=[illegible].4[illegible]5*PHT*B/10[illegible].
C           CALCULATION OF SYSTEM PERFORMANCE AND SYSTEM COSTS
C     MEMBRANE COSTS
C     MEMBRANE AREA REQUIRED , SQUARE FEET
      AREA=PROD(I)/WP(J)
      CMEMC=CMEM*AREA
      CMOPC=CMOP*AREA*2
C     CONTRIBUTION OF MEMBRANE TO WATER COST,CENT/1000. GALLONS
      CMEMCX=(CMEMC*(AMT + CINS)+CMOPC)*100000./(360.*PROD(I))
      CMOPCX=CMOPC*100000./(360.*PROD(I))
C      OPERATING COST FOR PUMP AND DRIVER IS (2/3 CPMP)+(25CENTS/OPER. HOUR)
C     TOTAL OPER. COST=COPP+COPP1
C     COPP1 BASED ON 30 YEARS,360 DAYS/YEAR,24H/DAY
C     COPP1 INCLUDES MAINT.+OPER. OF DRIVER+GEAR+PUMP
C     OPERATING COST FOR POWER RECOVERY SYSTEM IS (2/3 CHT)+(10CENTS/OPER. HOUR)
C     TOTAL OPERATING COST=COPHT+COPHT1
C     COPHT1 BASED ON 30 YEARS,360 DAYS/YEAR,24H/DAY
C     COPHT1 INCLUDES MAINT.+OPER. OF HYD. TURB. GEAR+ELEC.GEN.
      WRITE(6,1[illegible]1)PROD(I),P(J),R(K),PL(L)
      WRITE(6,1[illegible]8)WP(J),CMEMC,CMEMCX


C     SEA WATER PUMP CALCULATIONS

C     USE RECIPROCATING PUMPS FOR PLANT OUTPUT OF 100000 GPD OR LESS RPM=500
C     USE CENTRIFUGAL PUMPS IN ALL OTHER CASES,RPM=3600
C      FOR RECIPROCATING PUMPS USE BELTS AND NO GEARS
      IF(GPMP-2[illegible].)251,251,252
  251 SP(I)=500.
      IP=1
      EP=[illegible].9
      CPMP=16[illegible].
      GO TO 299
  252 SP(I)=36[illegible].
      IP=1
      IF(GPMP-2[illegible]00.)253,253,254
  253 EP=[illegible].85
      CPMP=25[illegible].
      GO TO 299
  254 EP= .[illegible]5
      CPMP=3[illegible].
  299 HPP=HP/EP
  219 CPMPX=CPMP*(AMT+C[illegible])*1[illegible]0[illegible]./(36[illegible].*PROD(I))
      GO TO 297
```

```
  297  COPP=CPMP*2./3.
       COPP1=.25*30.*360.*24.
       TCOP=COPP+COPP1
C     CONTRIBUTION OF PUMP AND DRIVER OPER. COST TO WATER COST,CENT/1000.GALLON
       COPX=(COPP*(AMT+CINS)+COPP1/30.)*100000./(360.*PROD(I))
       WRITE(6,110)
       WRITE(6,106)
       WRITE(6,107) HDP,GPMP,SP(IP),LP,HPP,CPMP,CPMPX
      IPRINT=0

C     HYDRAULIC TURBINE CALCULATIONS
C       HYDRAULIC TURBINE SPEEDS DEPEND ON FLOW RATE
C      3600 RPM FOR GPM LESS THAN 200
C      1800 RPM FOR GPM BETWEEN 200 AND 1500
C      1200 RPM FOR GPM BETWEEN 1500 AND 3500
C      900 RPM FOR GPM GREATER THAN 3500

      ET=0.85
C     TURBINE HORSEPOWER DEFINED AS EQUAL TO OUTPUT HORSEPOWER
       HPT=HPHT*ET
       IF(GPMT-200.) 2,3,3
    2  ST(4)=3600.
       IT=4
       GO TO 19
    3 IF(GPMT-1500.) 6,7,7
    6  ST(3)=1800.
       IT=3
       GO TO 19
    7  IF(GPMT-3500.) 8,9,9
    8  ST(2)=1200.
       IT=2
       GO TO 19
    9  ST(1)=900.
       IT=1
   19  CHT=13300.*(HPT/(HDHT)**0.5)**0.398
C     CONTRIBUTION OF HYDRAULIC TURBINE TO WATER COST,CENT/1000 GALLONS
   20  CHTX=CHT*(AMT + CINS)*100000./(360.*PROD(I))
   21  COPHT=CHT*2./3.
       COPHT1=.10*30.*360.*24.
       TCOHT=COPHT+COPHT1
C     CONTRIBUTION OF POWER RECOVERY OPER. COST TO WATER COST,CENT/1000.GALLONS
       COHTX=(COPHT*(AMT+CINS)+COPHT1/30.)*100000./(360.*PROD(I))
       WRITE(6,111)
       WRITE(6,106)
        WRITE(6,107)HDHT,GPMT,ST(IT),LT,HPT,CHT,CHTX
       WRITE(6,102)
       WRITE (6,104)


C     ELECTRIC GENERATOR CALCULATIONS
C     GENERATOR SPEEDS [illegible] , RPM
       IF(PLTC-1.)[illegible]
         S(1)=360.
       SG(2)=450.
       SG(3)=600.
       SG(4)=900.
       SG(5)=1200.
       SG(6)=1800.
        [illegible] IG=1,6
C     GENERATOR [illegible]
         = .8
       [illegible]
```

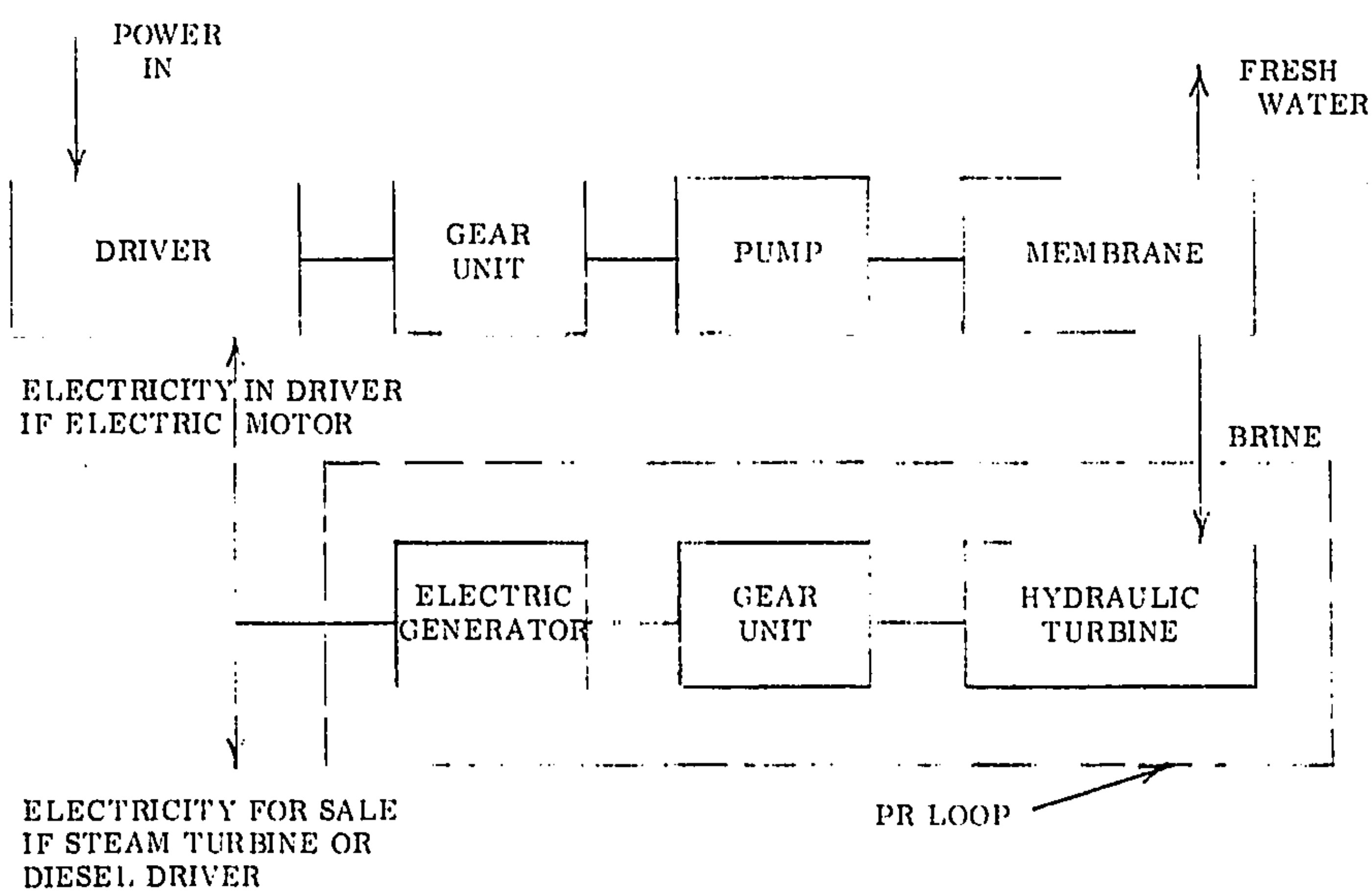
```
      GO TO(32,35,38,41,44,47),IG
C     GENERATOR SPEED = 360RPM
   32 IF(HPG-900./.7457)33,33,34
   33 CEG=546.40*((HPG*.7457)**.612)
      GO TO 52
   34 CEG=140.06*((HPG*.7457)**.812)
      GO TO 52
C     GENERATOR SPEED = 400RPM
   35 IF(HP-600./.7457)36,36,37
   36 CEG=568.18*((HPG*.7457)**.590)
      GO TO 52
   37 CEG=140.06*((HPG*.7457)**.810)
      GO TO 52
C     GENERATOR SPEED = 600RPM
   38 IF(HPG-400./.7457)39,39,40
   39 CEG=440.53*((HPG*.7457)**.586)
      GO TO 52
   40 CEG=230.41*((HPG*.7457)**.840)
      GO TO 52
C     GENERATOR SPEED = 900RPM
   41 IF(HPG-500./.7457)42,42,43
   42 CEG=124.38*((HPG*.7457)**.781)
      GO TO 52
   43 CEG=84.75*((HPG*.7457)**.841)
      GO TO 52
C     GENERATOR SPEED = 1200RPM
   44 IF(HPG-5000./.7457)45,45,46
   45 CEG=93.28*((HPG*.7457)**.815 )
      GO TO 52
   46 CEG=33.11*((HPG*.7457)**.936 )
      GO TO 52
C     GENERATOR SPEED = 1800RPM
   47 IF(HPG-350./.7457)48,48,49
   48 CEG=272.48*((HPG*.7457)**.592)
      GO TO 52
   49 IF(HPG-600./.7457)50,50,51
   50 CEG=19.76*((HPG*.7457)**.855)
      GO TO 52
   51 CEG=35.71*((HPG*.7457)**.986)
C     CONTRIBUTION OF ELECTRIC GENERATOR TO WATER COST, CENT/1000 GALLONS
   52 CEGX=CEG*(AMT+CINS)*10.000./(360.*PROD(I))
      GO TO 505
  504 CEG=0.0
      CEGX=0.0


C     ELECTRIC MOTOR CALCULATIONS - SQUIRREL CAGE
C     MOTOR SPEEDS INVESTIGATED , RPM
  505 SM(1)=400.
      SM(2)=1800.
      SM(3)=3600.
      DO 5   IM=1,3
C     ELECTRIC MOTOR HP DEFINED AS EQUAL TO OUTPUT HP
      EM=0.95
      IF(PLTC.EQ.1.0) GO TO 506
C     ELECTRIC MOTOR CALCULATIONS-SQUIRREL CAGE-WITHOUT RECOVERY
      HPM1=HPP/EM
      GO TO 507
C     ELECTRIC MOTOR WITH GEAR COUPLED TURBINE
C     CAPITAL COST
  506 HPM1=(HPP-HPT)/EM
  507 GO TO (15,16,17),IM
   15 IF(HPM1-3500.)18,18,22
   18 CMSC1=102.3*(HPM1**0.8 )
      GO TO 59
```

```
   22  CMSC1=20.8*HPM1
       GO TO 59
   16  CMSC1=10.8*HPM1
       GO TO 59
   17  CMSC1=14.0*HPM1
C      CONTRIBUTION OF ELECTRIC MOTOR TO WATER COST/1000 GALLONS
   59  CMSC1X=CMSC1*(AMT+CINS)*100000./(360.*PROD(I))
C      POWER COST , CENT/1000 GALLONS
       CPWR1X=CELE*HPM1*24.*100000./(PROD(I))*.7457
       HPM=(HPP-HPG)/EM
C      POWER COST, CENT/1000 GALLONS-WITH POWER RECOVERY
       CPWRX=CELE*HPM*24.*100000./(PROD(I))*.7457
       IF(PLTC-1.0)519,520,520
  520  CPWRX=CPWR1X
       CPWR1X=CELE*(HPP/EM)*24.*100000./(PROD(I))*.7457
       HPM2=HPP/EM
       GO TO (25,26,27),IM
   25  IF(HPM2-3500.)28,28,29
   28  CMSC2=182.3*(HPM2**0.809)
       GO TO 58
   29  CMSC2=20.8*HPM2
       GO TO 58
   26  CMSC2=10.3*HPM2
       GO TO 58
   27  CMSC2=14.0*HPM2
   58  CONTINUE


C            GEAR CALCULATIONS
C            MOTOR-PUMP GEARS
C      POWER TRANSMITTED THROUGH GEAR
  519  HPGMP=HPM1
       IF(SP(1)-500.)61,61,112
C      GEAR CAPITAL COST
  112  IF(SM(IM)-SP(IP)) 60,61,62
   61  CGRMP=0.
       GO TO 63
   60  CGRMP=CGR(HPGMP,SP(IP)/SM(IM),SP(IP))
       GO TO 63
   62  CGRMP=CGR(HPGMP,SM(IM)/SP(IP),SM(IM))
C      CONTRIBUTION OF GEAR TO WATER COST, CENT/1000 GALLONS
   63  CGRMPX=CGRMP*(AMT +*CINS )*100000./(360.*PROD(I))
       IF(PLTC.EQ.1.0) GO TO 508



C            HYDRAULIC TURBINE- ELECTRIC GENERATOR
C      POWER TRANSMITTED THROUGH GEAR
       HPGTG=HPHT*FT
C      GEAR CAPITAL COST
       IF(ST(IT)-SG(IG)) 68,69,70
   69  CGRTG=0.0
       GO TO 71
   70  CGRTG=CGR(HPGTG,ST(IT)/SG(IG),ST(IT))
       GO TO 71
   68  CGRTG=CGR(HPGTG,SG(IG)/ST(IT),SG(IG))
C      CONTRIBUTION OF GEAR TO WATER CO T ,CENT/10   [illegible]LLON
   71  CGRTGX=CGRTG*(AMT + CINS )*10    ./(36 .*PROD(I))
       GO TO 509

C            HYDRAULIC TURBINE-PUMP GEAR
  508  HPGTP=HPT
C      POWER TRANSMITTED THROUGH GEAR
```

```
C      GEAR CAPITAL COST
        IF(ST(II)-SP(IP))88,89,90
   89  CGRTP=0.0
        GO TO 91
   90  CGRTP=CGR(HPGTP,ST(II)/SP(IP), T(II))
        GO TO 91
   88  CGRTP=CGR(HPGTP,SP(IP)/ST(II), T(IP))
C      CONTRIBUTION OF GEAR TO WATER COST, CENT/1000GAL
   91  CGRTPX=CGRTP*(AMT+CINS)*100000./(360.*PROD(I))
        CGRTG=CGRTP
        CGRTGX=CGRTPX


C      PLANT FIRST COSTS
  509  FCOST=CPMP+CHT+CMEMC+CGRTG+CMSC1+CEG+CGRMP
        FCOST1=CPMP+CMEMC+CMSC1+CGRMP
        IF(PLTC.EQ.1.0)FCOST=CPMP+CMEMC+CMSC2+CGRMP
C          TOTAL WATER COST
C      CAPITAL WATER COSTS IN CENT/1000 GALLONS WITH POWER RECOVERY
        CAPX=FCOST*(AMT+CINS)*100000./(360.*PROD(I))
C      CAPITAL WATER COSTS WITHOUT POWER RECOVERY
        CAP1X=FCOST1*(AMT+CINS)*100000./(360.*PROD(I))
C      OPERATING COSTS AND POWER COSTS IN CENTS/1000 GALLONS
        OPCX=COPX+COHTX+CPWRX+CMOPCX
C      OPERATING COSTS AND POWER COSTS WITHOUT POWER RECOVERY
        OPC1X=COPX+CPWR1X+CMOPCX
C      TOTAL WATER COSTS IN CENTS/1000 GALLONS WITH POWER RECOVERY
        CTOTX=CAPX+OPCX
C      TOTAL WATER COSTS IN CENTS/1000 GALLONS WITHOUT POWER RECOVERY
        CTOT1X=CAP1X+OPC1X
C      SAVINGS OR LOSSES DUE TO POWER RECOVERY SYSTEM
        DELTA=CTOT1X-CTOTX
C      IF DELTA IS POSITIVE POWER RECOVERY SYSTEM WILL BE PROFITABLE
C      IF DELTA IS LESS THAN OR EQUAL TO ZERO,THEN A POWER RECOVERY SYSTEM
C      SHOULD NOT BE RECOMMENDED


       IPRINT=IPRINT + 1
        IF(IPRINT -36) 499,499,498
  498 WRITE(6,100)
       WRITE(6,102)
        WRITE (6,104)
       IPRINT = 0
  499  WRITE(6,103)SM(IM),SG(IG),TCOP,TCOHT,CGRTP,CGRTG,CMSC1,CEG,COPX,C
      1OHTX,CMSC1X,CEGX,CTOTX,CTOT1X,DELTA
        IF(SM(IM).EQ.3600..AND.SG(IG).EQ.1800.)WRITE(6,109)CPWRX,CPWR1X

        IF(PLTC.EQ.1.AND.SM(IM).EQ.3600.)WRITE (6,109)CPWRX,CPWR1X
  500  CONTINUE
        PLTC=PLTC+1.0
        IF(PLTC-1.0)501,501,502
  502  CONTINUE
        STOP
        END
          SCOPE
*LOAD
*RUN,2,99999,,1
**
```



(a)



(b)

Figure 8.1.1 General Power Recovery Systems for Reverse Osmosis Desalination Plants

PROGRAM PLANT FLOW CHART

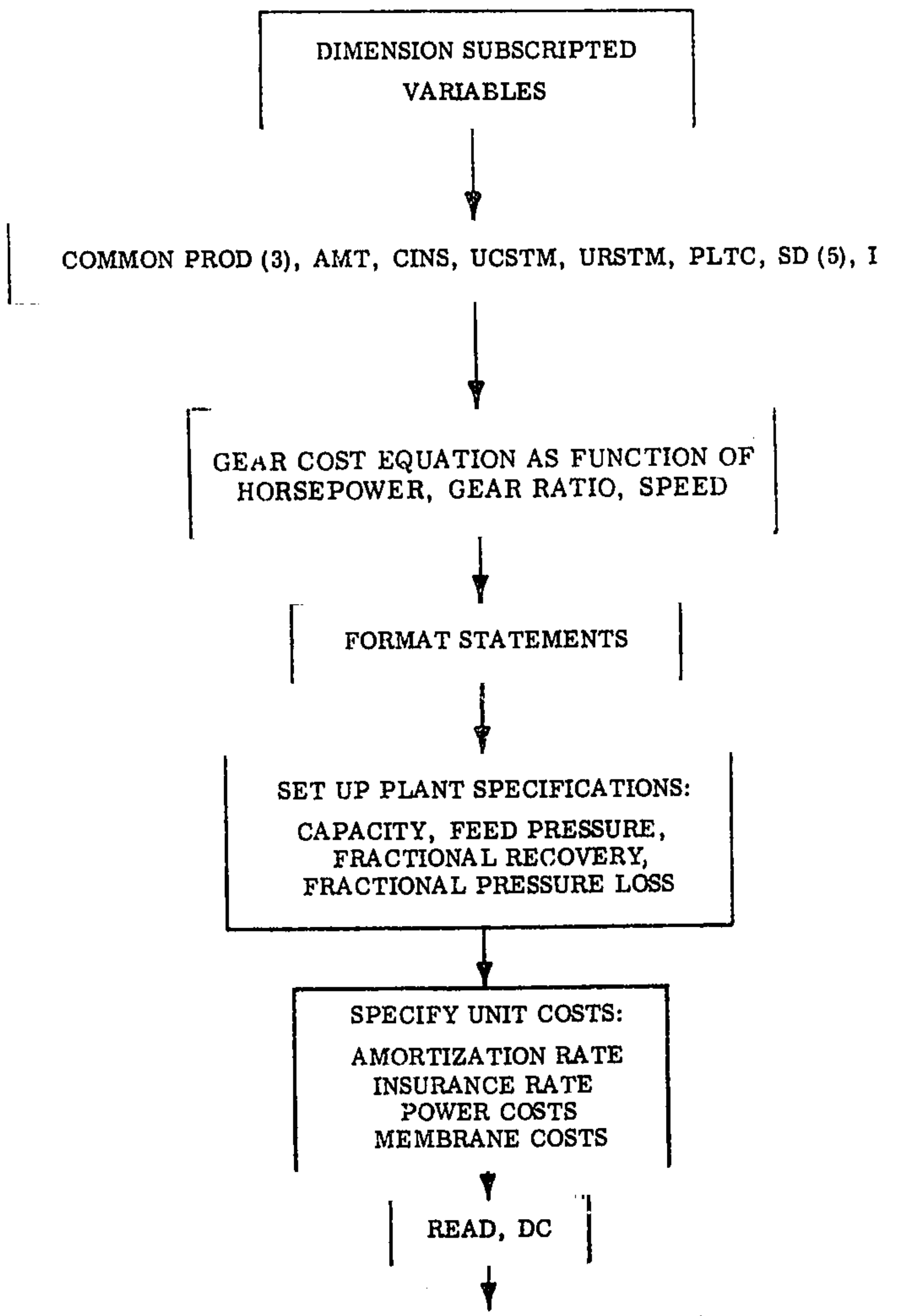


Figure 8.2.1 Program Plant Flow Chart

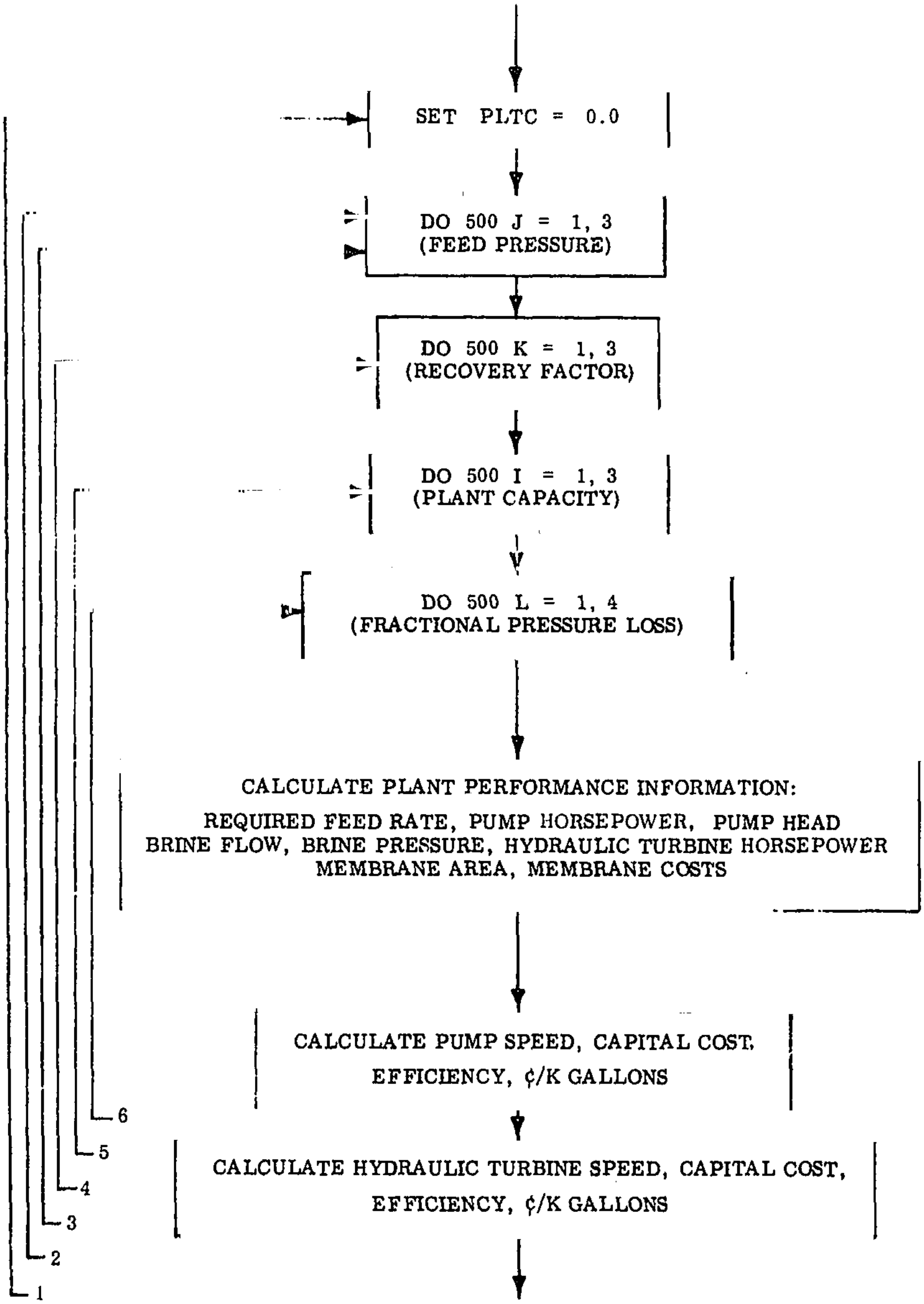


Figure 8.2.1 Program Plant Flow Chart (Continued)

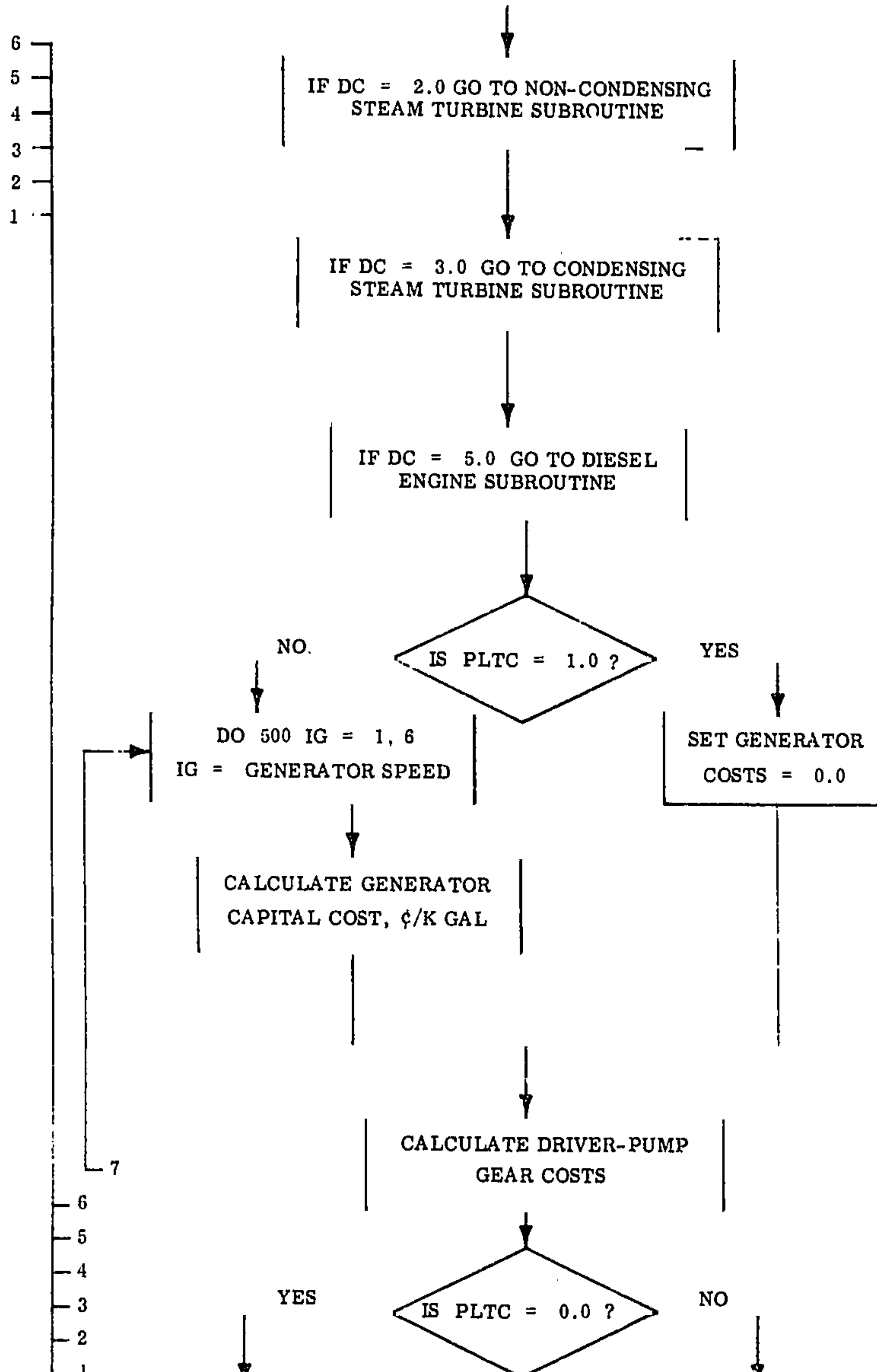


Figure 8.2.1 Program Plant Flow Chart (Continued)

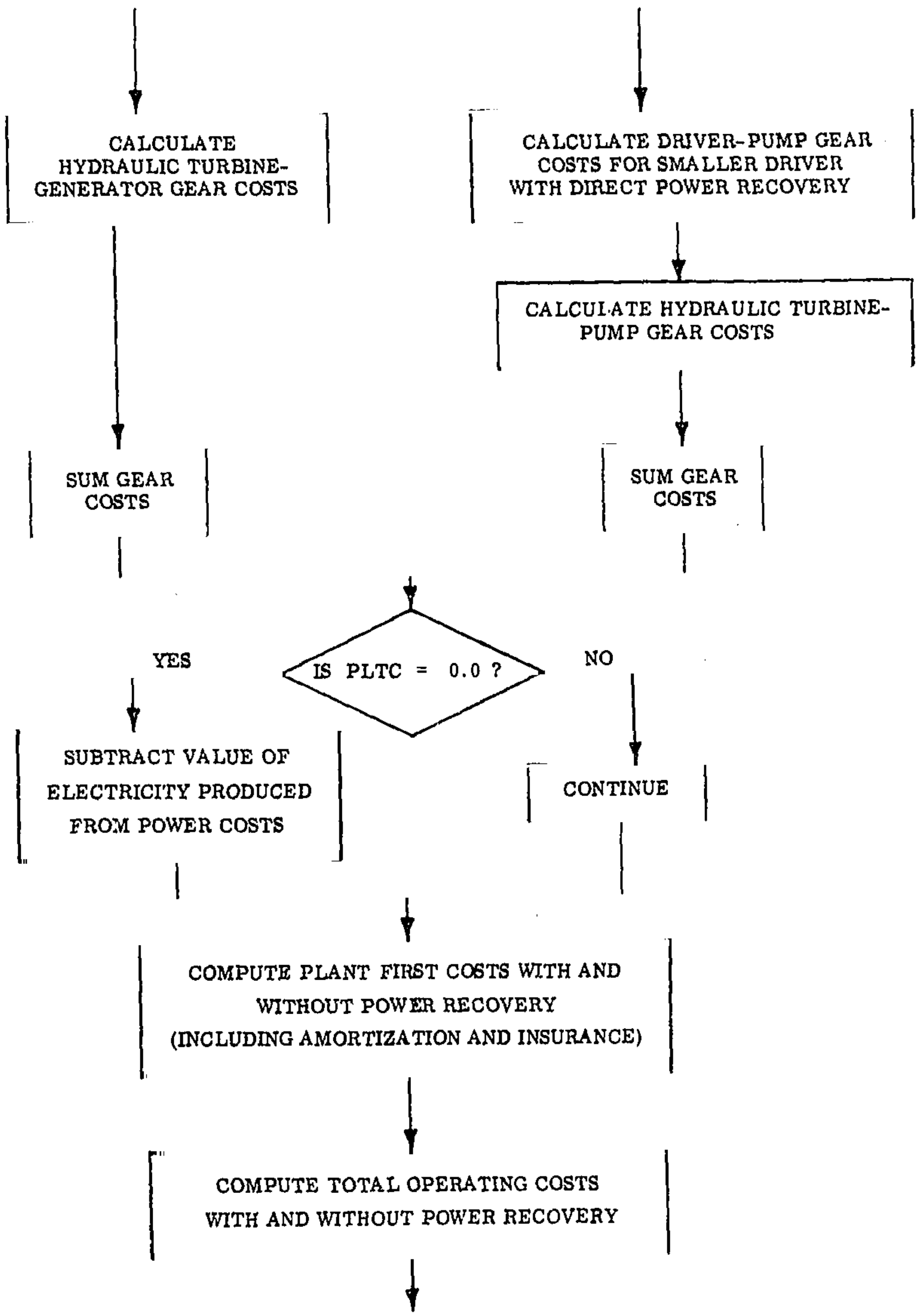


Figure 8.2.1 Program Plant Flow Chart (Continued)

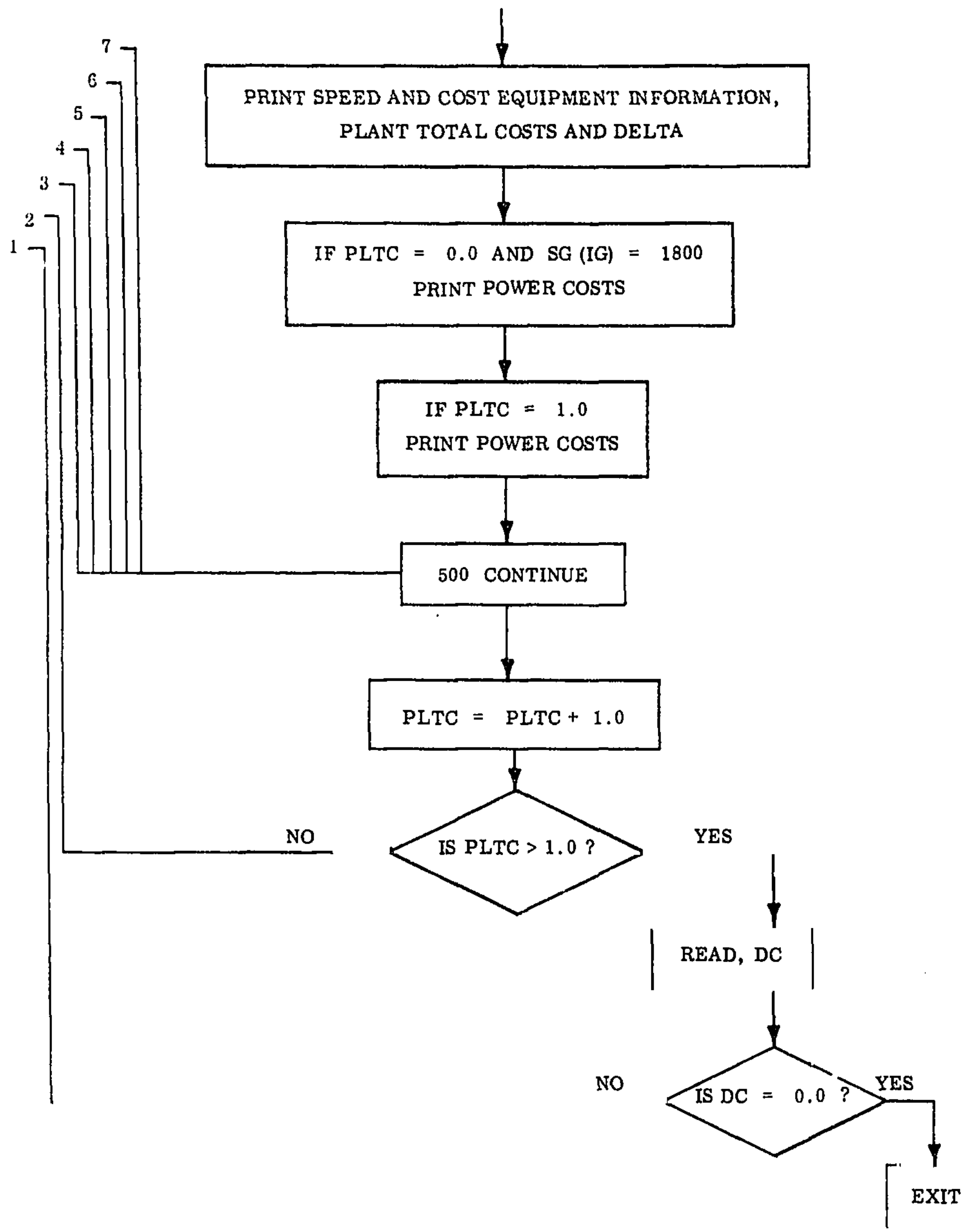


Figure 8.2.1 Program Plant Flow Chart (Continued)

GPD= 100000 PSI= 800 RECOVERY=0.8 FRACTION P-LOSS=0.00

MEMBRANE THRUPUT,GPD/FT2= 20 CAPITAL COST=$ 1250 CENT/1000 GAL= 7.1

PUMP DATA

HEAD,FT	FLOW RATE,GPM	SPEED,RPM	EFFICIENCY	HORSEPOWER	CAPITAL COST,$	CENT/1000 GAL
1766.3	86.6	500	0.90	44.2	2800	0.4

HYDRAULIC TURBINE DATA

HEAD,FT	FLOW RATE,GPM	SPEED,RPM	EFFICIENCY	HORSEPOWER	CAPITAL COST,$	CENT/1000 GAL
1766.3	17.4	3500	0.85	6.8	4976	0.8

SPEEDS,RPM		** OPERATING AND CAPITAL COSTS,DOLLARS						** CONTRIBUTIONS TO WATER COST, CENTS/1000GAL						
MOT	GEN	PUMP	TURB	GEAR1	GEAR2	MOTOR	GEN	PUMP	TURB	MOTR	GEN	TOT1	TOT2	DELTA
400	360	66667	29237	0	163	2386	1282	6.3	2.9	0.4	0.2	23.5	20.4	-3.1
1800	360	66667	29237	0	163	530	1282	6.3	2.9	0.1	0.2	23.2	20.1	-3.1
3600	360	66667	29237	0	163	687	1282	6.3	2.9	0.1	0.2	23.2	20.1	-3.1
400	400	66667	29237	0	147	2386	1293	6.3	2.9	0.4	0.2	23.5	20.4	-3.1
1800	400	66667	29237	0	147	530	1293	6.3	2.9	0.1	0.2	23.2	20.1	-3.1
3600	400	66667	29237	0	147	687	1293	6.3	2.9	0.1	0.2	23.2	20.1	-3.1
400	600	66667	29237	0	101	2386	997	6.3	2.9	0.4	0.2	23.5	20.4	-3.1
1800	600	66667	29237	0	101	530	997	6.3	2.9	0.1	0.2	23.2	20.1	-3.1
3600	600	66667	29237	0	101	687	997	6.3	2.9	0.1	0.2	23.2	20.1	-3.1
400	900	66667	29237	0	71	2386	369	6.3	2.9	0.4	0.1	23.4	20.4	-3.0
1800	900	66667	29237	0	71	530	369	6.3	2.9	0.1	0.1	23.1	20.1	-3.0
3600	900	66667	29237	0	71	687	369	6.3	2.9	0.1	0.1	23.1	20.1	-3.0
400	1200	66667	29237	0	56	2386	290	6.3	2.9	0.4	0.0	23.3	20.4	-3.0
1800	1200	66667	29237	0	56	530	290	6.3	2.9	0.1	0.0	23.1	20.1	-3.0
3600	1200	66667	29237	0	56	687	290	6.3	2.9	0.1	0.0	23.1	20.1	-3.0
400	1800	66667	29237	0	40	2386	622	6.3	2.9	0.4	0.1	23.4	20.4	-3.0
1800	1800	66667	29237	0	40	530	622	6.3	2.9	0.1	0.1	23.1	20.1	-3.0
3600	1800	66667	29237	0	40	687	622	6.3	2.9	0.1	0.1	23.1	20.1	-3.0

POWER (CENT/1000 GAL) WITH RECOV= 5.4 WITHOUT RECOV= 6.1

Figure 8.2.2 Program 4 Output - Electric Motor Driven Brackish Water Plant

```
GPD= 1000000     PSI= 600     RECOVERY=0.5     FRACTION P-LOSS=0.00

MEMBRANE     THRUPUT,GPD/FT2= 15     CAPITAL COST=$   16667     CENT/1000 GAL=  9.5
```

PUMP DATA

HEAD,FT	FLOW RATE,GPM	SPEED,RPM	EFFICIENCY	HORSEPOWER	CAPITAL COST,$	CENT/1000 GAL
1316.3	1388.9	3600	0.71	667.4	12124	0.2

HYDRAULIC TURBINE DATA

HEAD,FT	FLOW RATE,GPM	SPEED,RPM	EFFICIENCY	HORSEPOWER	CAPITAL COST,$	CENT/1000 GAL
1316.3	694.4	1800	0.85	201.4	20375	0.3

SPEEDS,RPM		** OPERATING AND CAPITAL COSTS,DOLLARS						** CONTRIBUTIONS TO WATER COST, CENTS/1000GAL						
MOT	GEN	PUMP	TURB	GEAR1	GEAR2	MOTOR	GEN	PUMP	TURB	MOTR	GEN	TOT1	TOT2	DELTA
400	-0	72883	39503	7414	865	16055	0	0.7	0.4	0.2	0.0	17.8	20.1	2.3
1800	-0	72883	39503	2030	865	5592	0	0.7	0.4	0.1	0.0	17.5	19.8	2.3
3600	-0	72883	39503	0	865	7249	0	0.7	0.4	0.1	0.0	17.5	19.8	2.3

```
POWER (CENT/1000 GAL)     WITH RECOV=  6.5     WITHOUT RECOV=  9.3
```

Figure 8.2.3 Program 1 Output - Electric Motor Driven Brackish Water Plant, No Electric Generator

Section 9

RESULTS OF THE COMPUTER CALCULATIONS

9.1 Contributions to Fresh Water Cost

The results obtained from the computer analyses are presented in this section. A series of curves have been prepared to show the contribution to the total cost of fresh water incurred by the acquisition, operation, and servicing of the pumping system equipment.

The particular costs that have been included in this study and which are represented as the "contribution to water cost" on these curves were described in Section 7. A complete list of all the separate cost items follows:

driver capital cost:
- electric motor price
- non-condensing steam turbine price
- condensing steam turbine price
- diesel engine price
- gear, between pump and driver, price

- pump price
- pump service and maintenance costs
- hydraulic turbine price
- hydraulic turbine service and maintenance costs
- price of gear between hydraulic turbine and pump, or price of gear between hydraulic turbine and generator
- electric generator price
- membrane initial cost
- membrane replacement and service cost
- power costs for the driver

A legend is provided before each set of curves explaining the symbols used. The shaded areas on the plots represent pumping systems that include an energy recovery system. The amount of energy recovered by the hydraulic turbine is influenced by the pressure drop in the brine as it flows from the pump through the membrane array to the turbine. In this investigation, pressure drop values of 0%, 5%, 10%, and 15% of the pump discharge pressure level were each used by the computer during the cost calculations for alternative pumping systems. Therefore, the upper (higher cost) boundary of the shaded area represents 15% pressure drop in the membrane array, while the lower (lower cost) boundary represents 0% pressure drop. The curves for 5% and 10% pressure drops fall within the shaded area.

When energy recovery is not included in the system, the total contribution of the pumping system to the water cost is represented by a single line rather than a band.

One set of figures (Figures 9.1.31 to 9.1.46) shows the individual contributions of the capital costs, operating costs, and driver power costs for the various plant arrangements. Each of these costs are separately represented by a line which is appropriately labeled (e.g., turbine operating, pump capital). Each cost is referenced to the baseline of the figure. This permits a direct comparison of the importance of each component of the "total contribution to the water cost".

9.2 Results of Optimization Calculations

9.2.1 Comparison of Various Plant Arrangements

An analysis of the figures given in the previous section shows that, for the 10^5 GPD plant size, the addition of an energy recovery turbine is not economically justified. Furthermore, the type of driver used (motor, steam turbine, or diesel engine) has an insignificant effect on the fresh water cost.

For the 10^6 GPD and 10^7 GPD plant sizes, the selection of the type of driver becomes important and energy recovery becomes economically justifiable. The plots show that the power costs for the 10^7 GPD plant size are ten times larger than the total capital cost of all the pumping system components, and seven times larger than the "operating" costs (service and maintenance). Therefore, the driver efficiency and the cost of the energy supply (electricity, steam, or fuel oil) become very important factors in the economy of the larger plants. (In the case of the 10^5 GPD plant, the power costs were only 2 times larger than capital costs, and 1/5 of the "operating" costs).

For the larger plant sizes, the power costs follow this pattern: steam turbines are less expensive than diesel engines, which are much less expensive than electric motors. The quantitative values follow:

- non-condensing steam turbine drive with energy recovery: 10¢/k gal
- condensing steam turbine drive with energy recovery: 11¢/k gal
- diesel engine drive with energy recovery: 14¢/k gal
- electric motor drive with energy recovery: 16¢/k gal

The savings achieved by the energy recovery turbine for the 10^6 GPD to 10^7 GPD plant size can be conveniently expressed as a percentage of the total cost contribution of the pumping system without energy recovery. This percentage was influenced by the type of driver used and the energy recovery arrangement but was not influenced by the membrane pressure level, recovery factor, brine-side pressure drop, or feedwater flow rate. The savings percentages were as follows:

Driver	Savings in % of total cost contribution without energy recovery
electric motor	10%
diesel engine	15%
steam turbine	20%

The savings values above are for an energy recovery system in which the hydraulic turbine drives an electrical generator. If the turbine is used to drive the pump directly or through a gear, the savings are increased by about 1 to 2% of the total cost for plants delivering 10^6 GPD of fresh water, and by about 1/2 % for plants delivering 10^7 GPD.

9.2.2 Influence of Pressure Level and Recovery Factor

Three parameters taken together can be used to represent each "standard" plant; these are the fresh water recovery factor, the membrane pressure level, and the fresh water production rate. Table 9.2.1, taken from Figures 9.1.1 to 9.1.18, shows the limiting values of these parameters at which energy recovery becomes economical for electric motor driven systems. (For the diesel engine and steam turbine driven systems, energy recovery is economical for all of the 10^6 and 10^7 GPD plants considered.)

Electric Generator			Hydraulic Turbine Direct Hook-up		
Recovery Factor	Pressure	Fresh Water Output	Recovery Factor	Pressure	Fresh Water Output
50%	1000 PSIA	10^5 GPD	50%	800 PSIA	10^5 GPD
70%	400 PSIA	10^6 GPD	70%	400 PSIA	10^6 GPD
80%	600 PSIA	10^6 GPD	80%	400 PSIA	10^6 GPD
80%	400 PSIA	10^7 GPD	80%	400 PSIA	10^7 GPD

Table 9.2.1 Combinations of Recovery Factor, System Pressure and Fresh Water Output at Which Energy Recovery Becomes Economical

The conclusions for electric motor driven systems are as follows:

- At a recovery factor of 50%, enough rejected brine leaves the membrane array to make energy recovery economically sound at all of the system pressures considered for plant sizes of 10^6 and 10^7 GPD. For the 10^5 GPD plant size, energy recovery is not economical if the system pressure is less than 1000 psi.

- At a recovery factor of 70%, a smaller amount of rejected brine is available to the energy recovery system. Energy recovery is not economical for 10^5 GPD plants but is economical for all of the 10^6 and 10^7 GPD plants considered.

- At a recovery factor of 80%, energy recovery is not economical for 10^5 GPD plants, and is economical for all of the 10^7 GPD plants considered. For 10^6 GPD plant size, the energy recovery is economical only for pressures of 600 psi or greater. when an electric generator is included in the energy recovery system.

- Energy recovery is always economically justified for the 1500 psi sea water desalination plants with flow rates of 10^5 GPD or greater.

		Figure No.
————————	Without Energy Recovery	9. 1. 1
— — — — —	With Electric Generator	to
— ——	With Direct H. T. Hook-up	9. 1. 18

Type of Driver

		Figure No.
	Electric Motor	9. 1. 19
— — —	Diesel Engine	to
....	Condensing Steam Turbine	
-	Non-Condensing Steam Turbine	9. 1. 30

From Figure 9. 1. 31 to Figure 9. 1. 46. all symbols are explained.

Explanation of Symbols Used in the Various Figures Describing the Contributions to Total Water Cost for Brackish Water Desalination Plants.

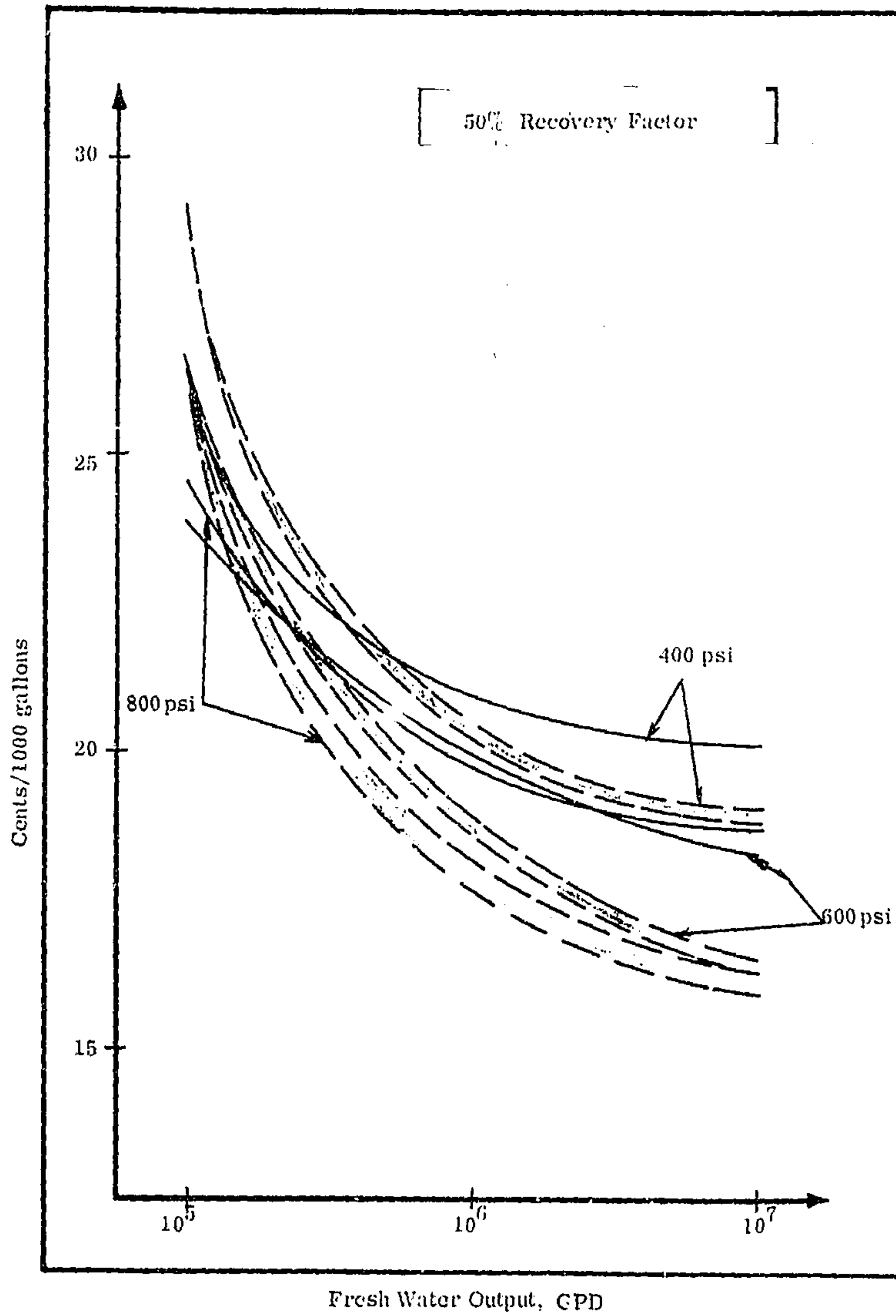


Figure 9. 1. 1 Contribution to Total Water Cost - Electric Motor Driven Plant with Electric Generator

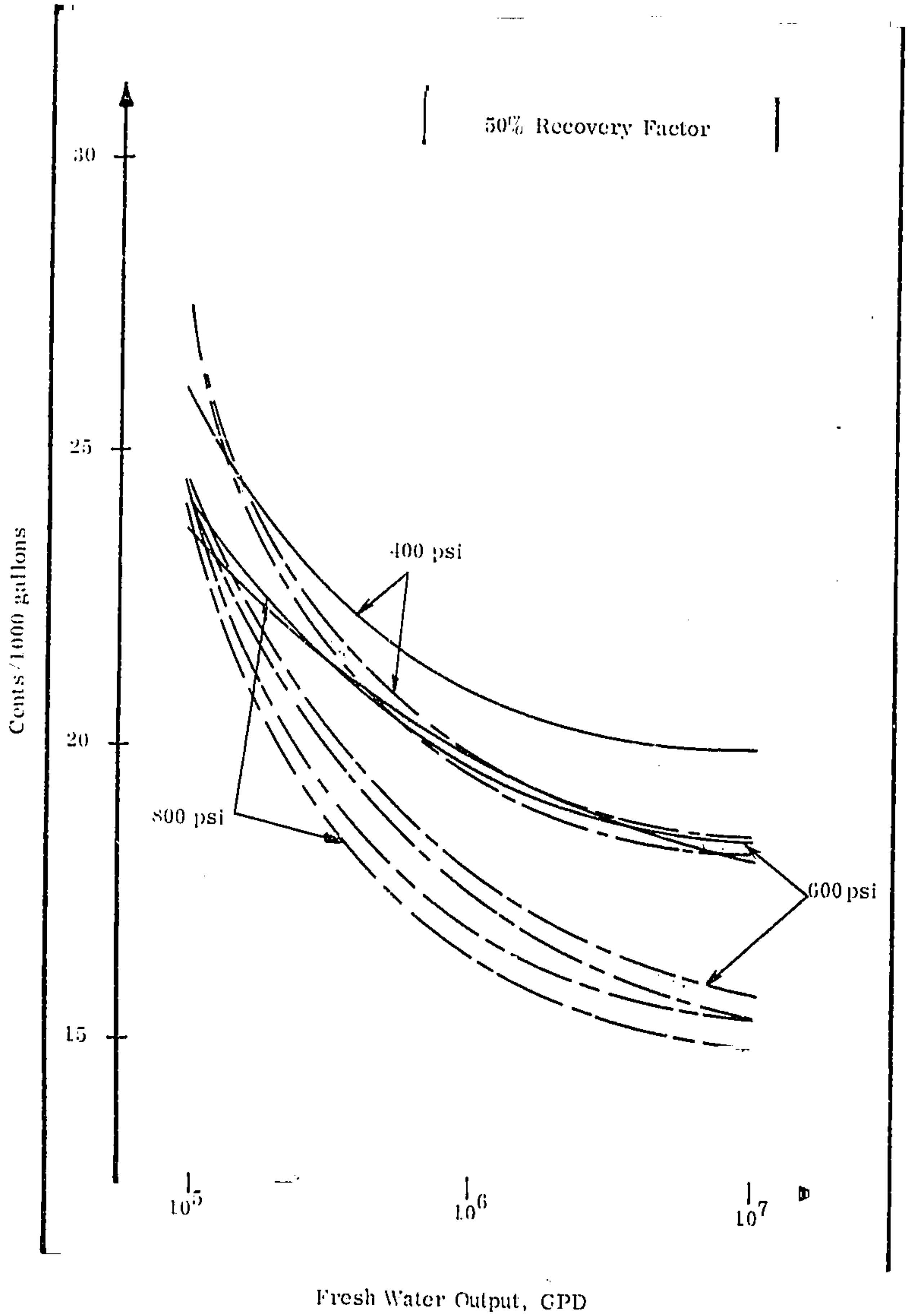


Figure 9.1.2 Contribution to Total Water Cost - Electric Motor Driven Plant with Direct Hydraulic Turbine Hook-up

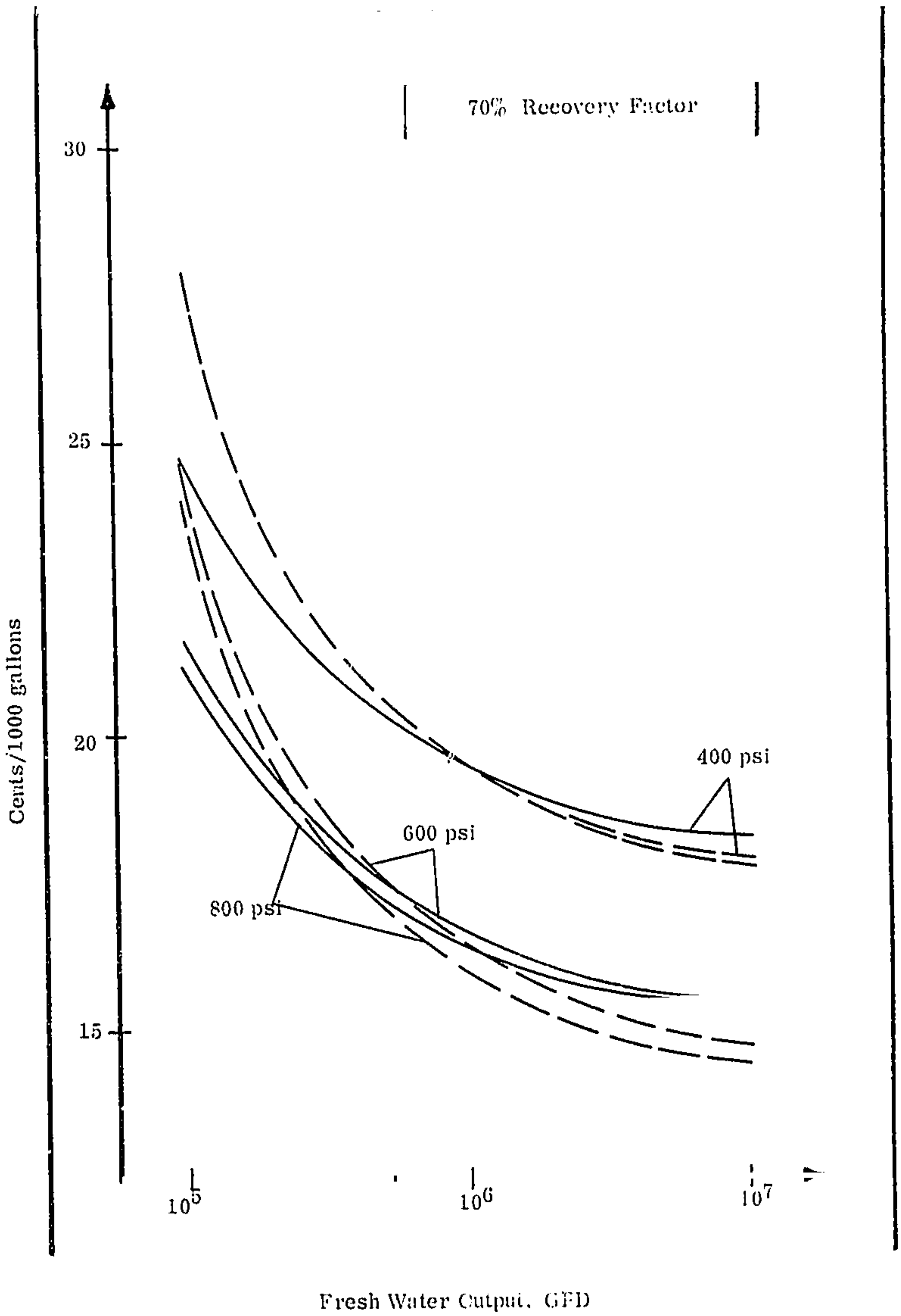


Figure 9. 1. 3 Contribution to Total Water Cost - Electric Motor Driven Plant with Electric Generator

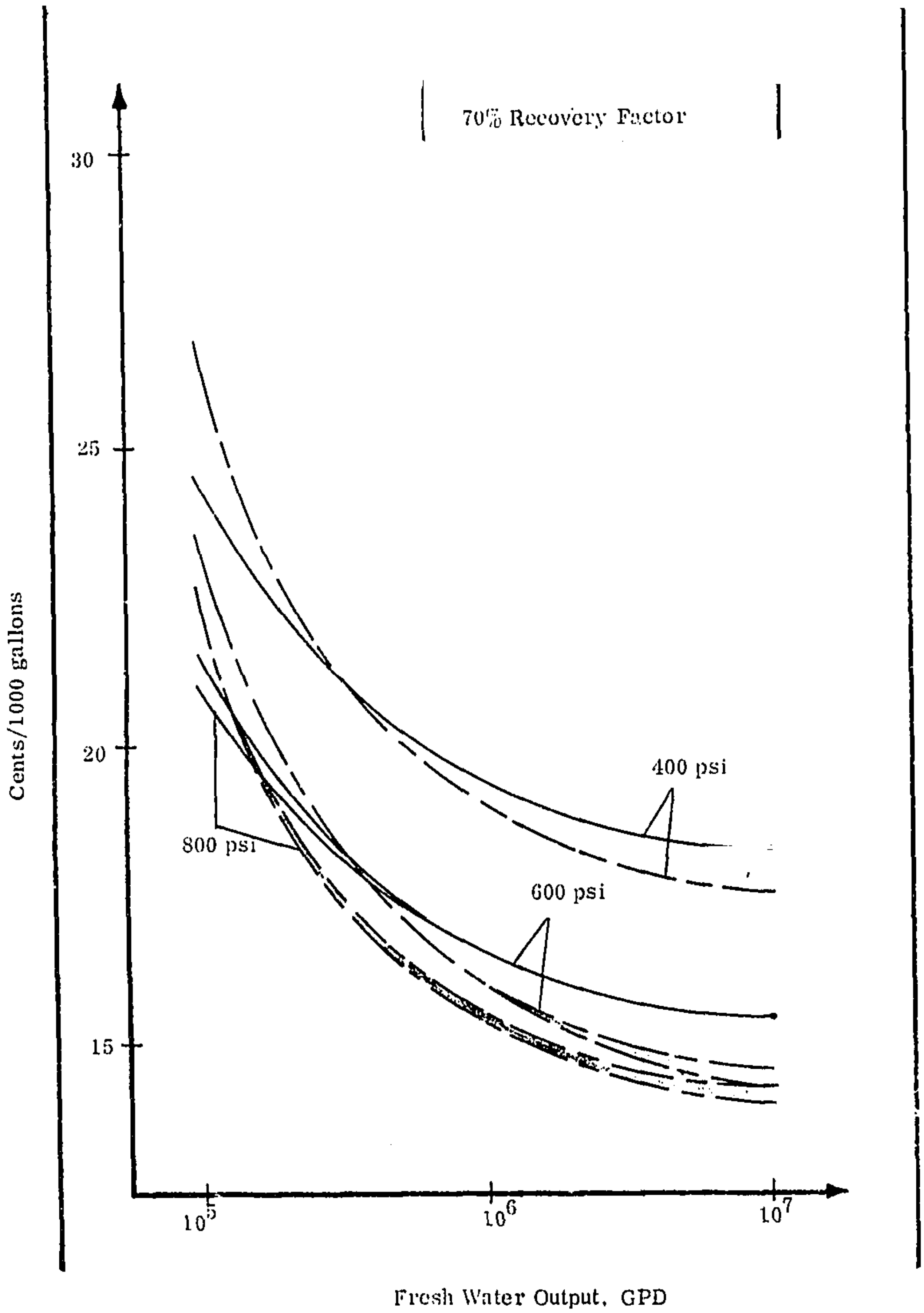


Figure 9.1.4 Contribution to Total Water Cost - Electric Motor Driven Plant with Direct Hydraulic Turbine Hook-up

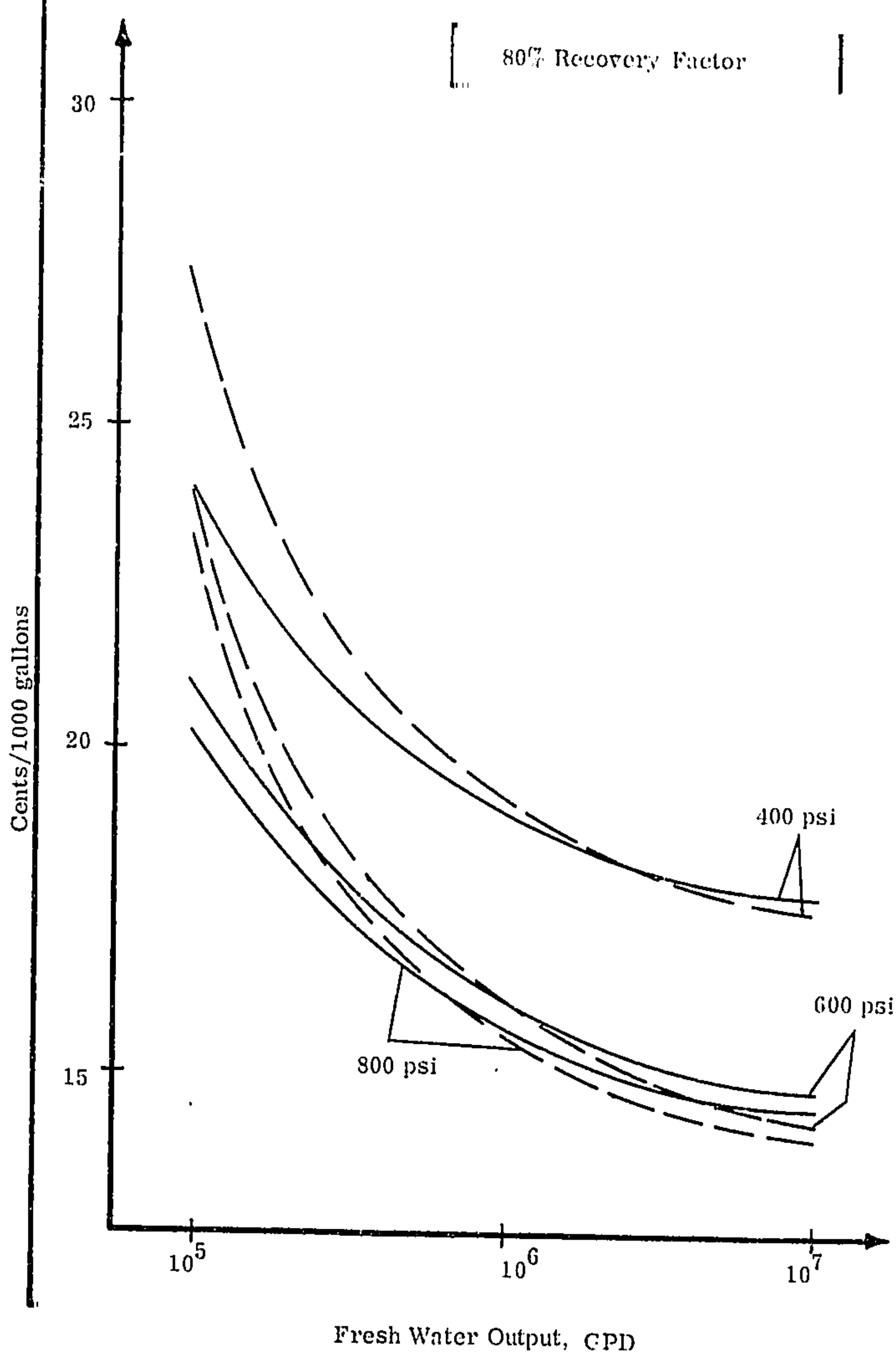


Figure 9.1.5 Contribution to Total Water Cost - Electric Motor Driven Plant with Electric Generator

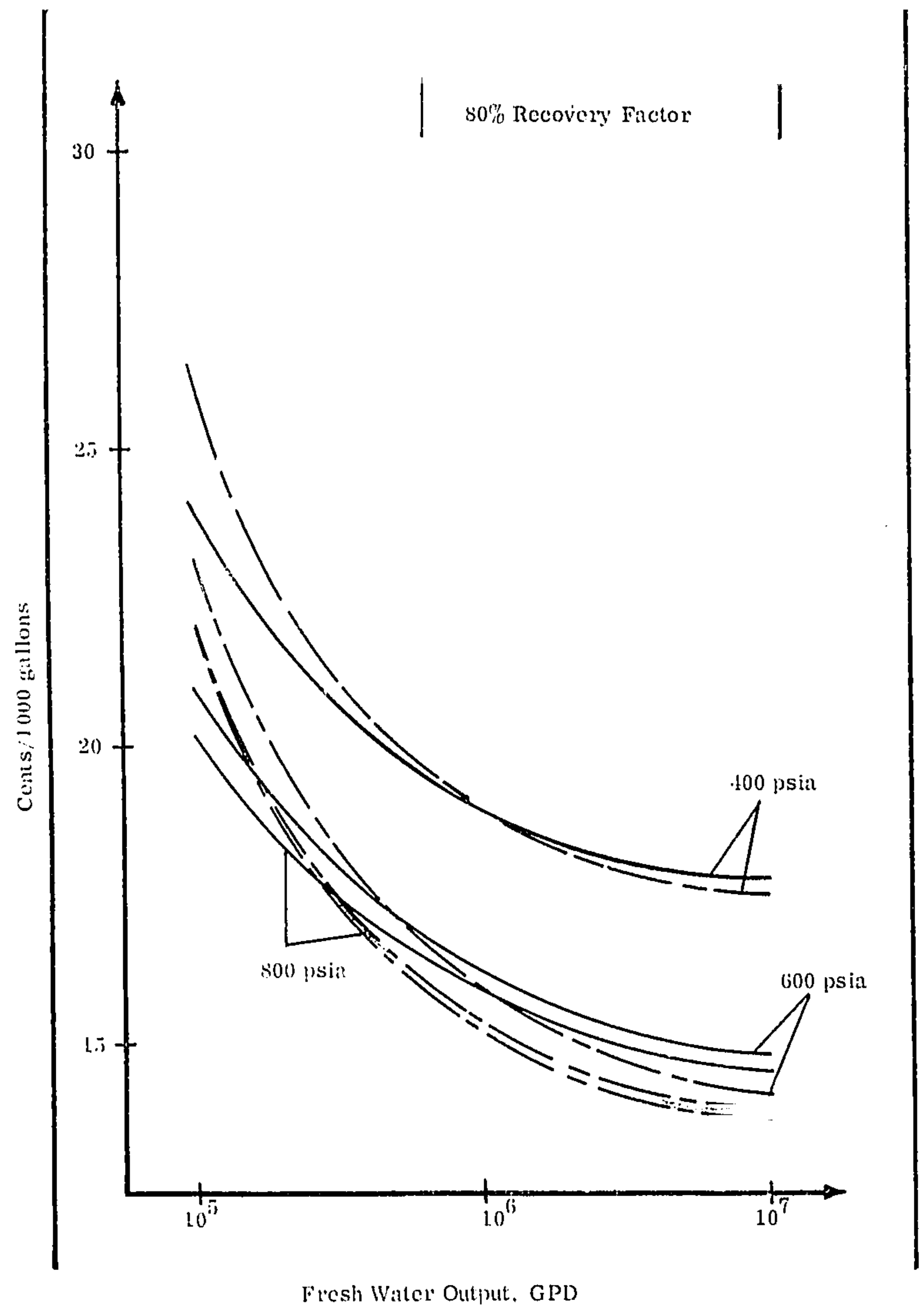


Figure 9.1.6 Contribution to Total Water Cost - Electric Motor Driven Plant with Direct Hydraulic Turbine Hook-up

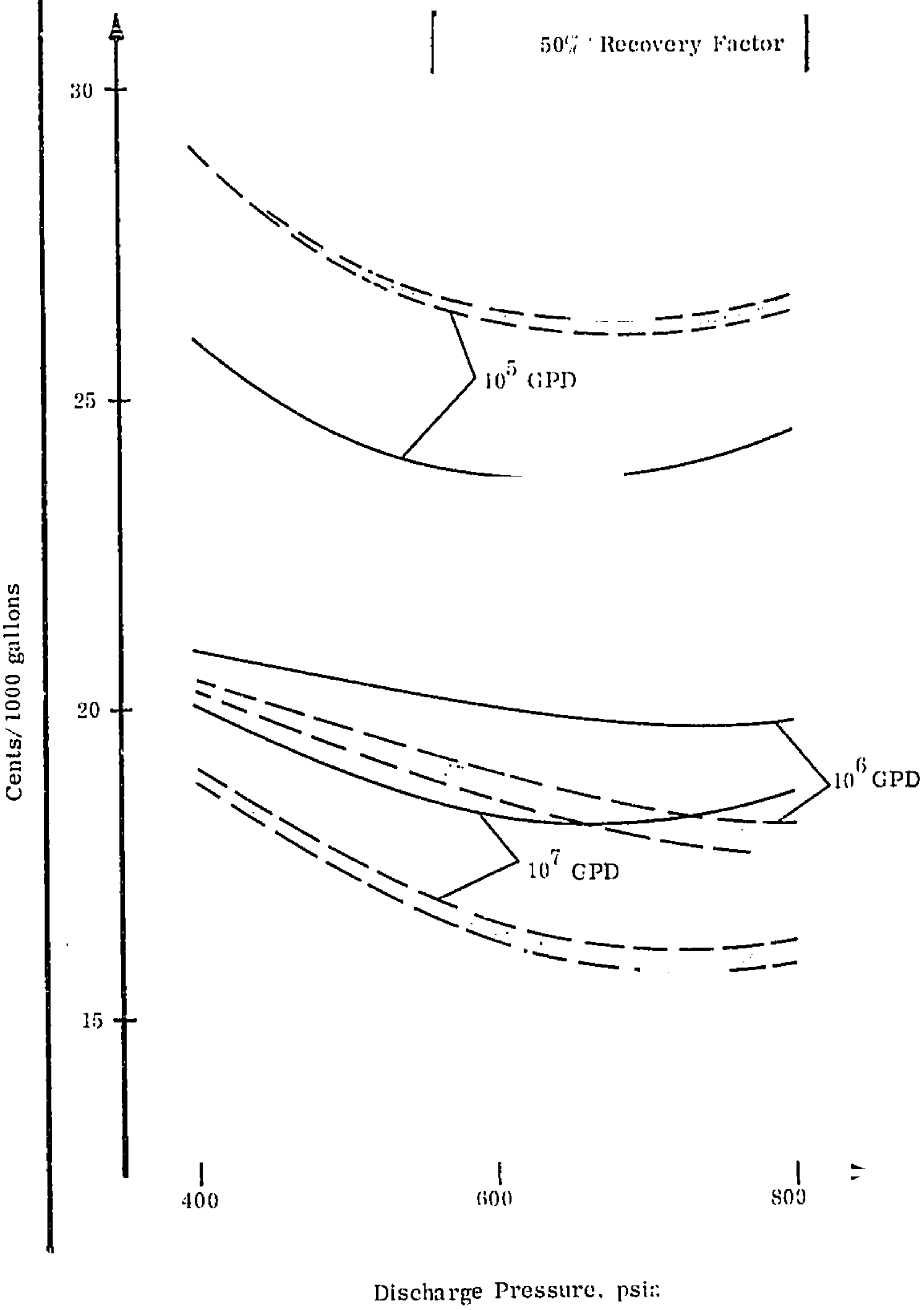
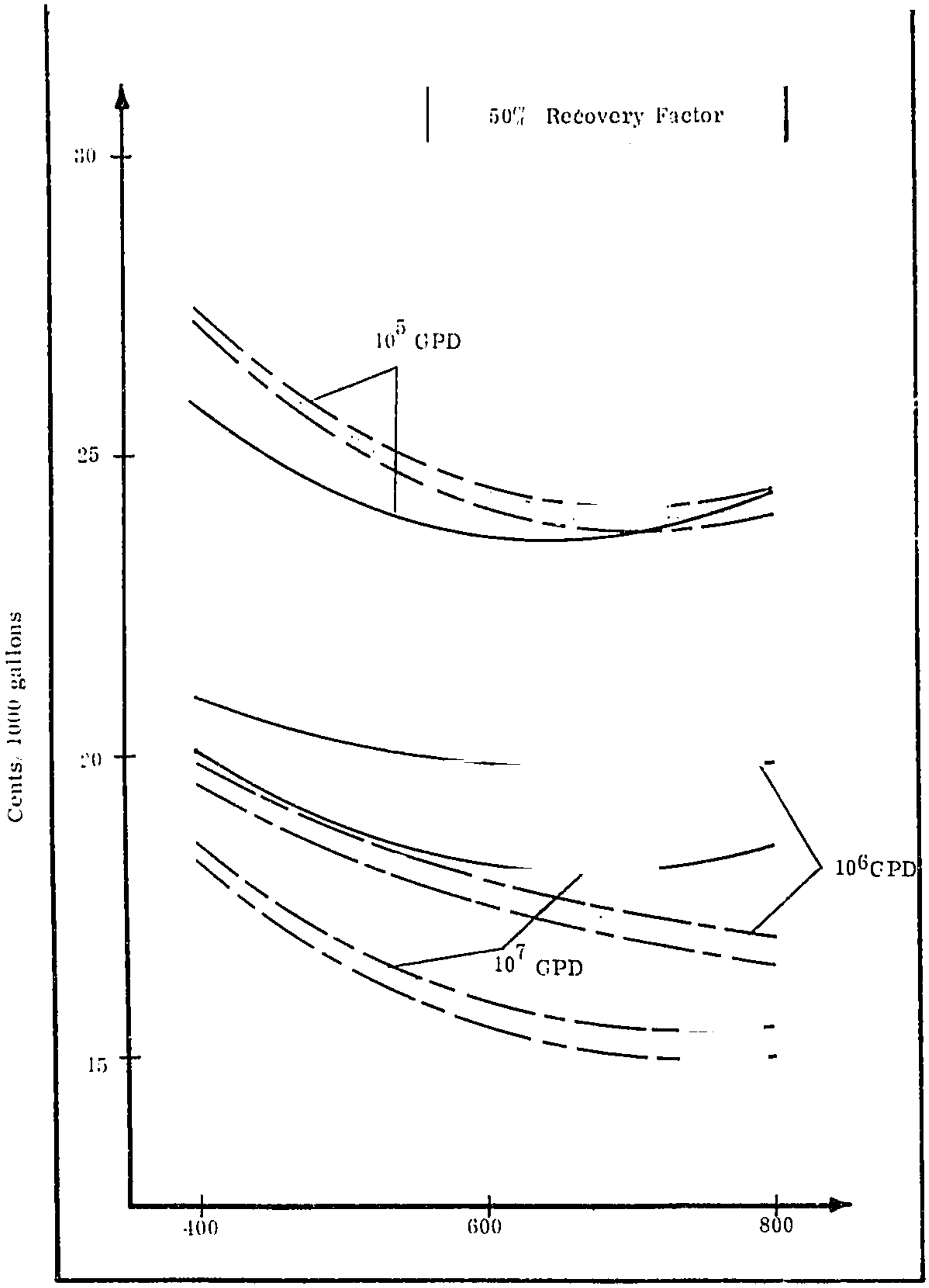


Figure 9. 1. 7 Contribution to Total Water Cost - Electric Motor Driven Plant with Electric Generator



Discharge Pressure, psia

Figure 9.1.8 Contribution to Total Water Cost - Electric Motor Driven Plant with Direct Hydraulic Turbine Hook-up

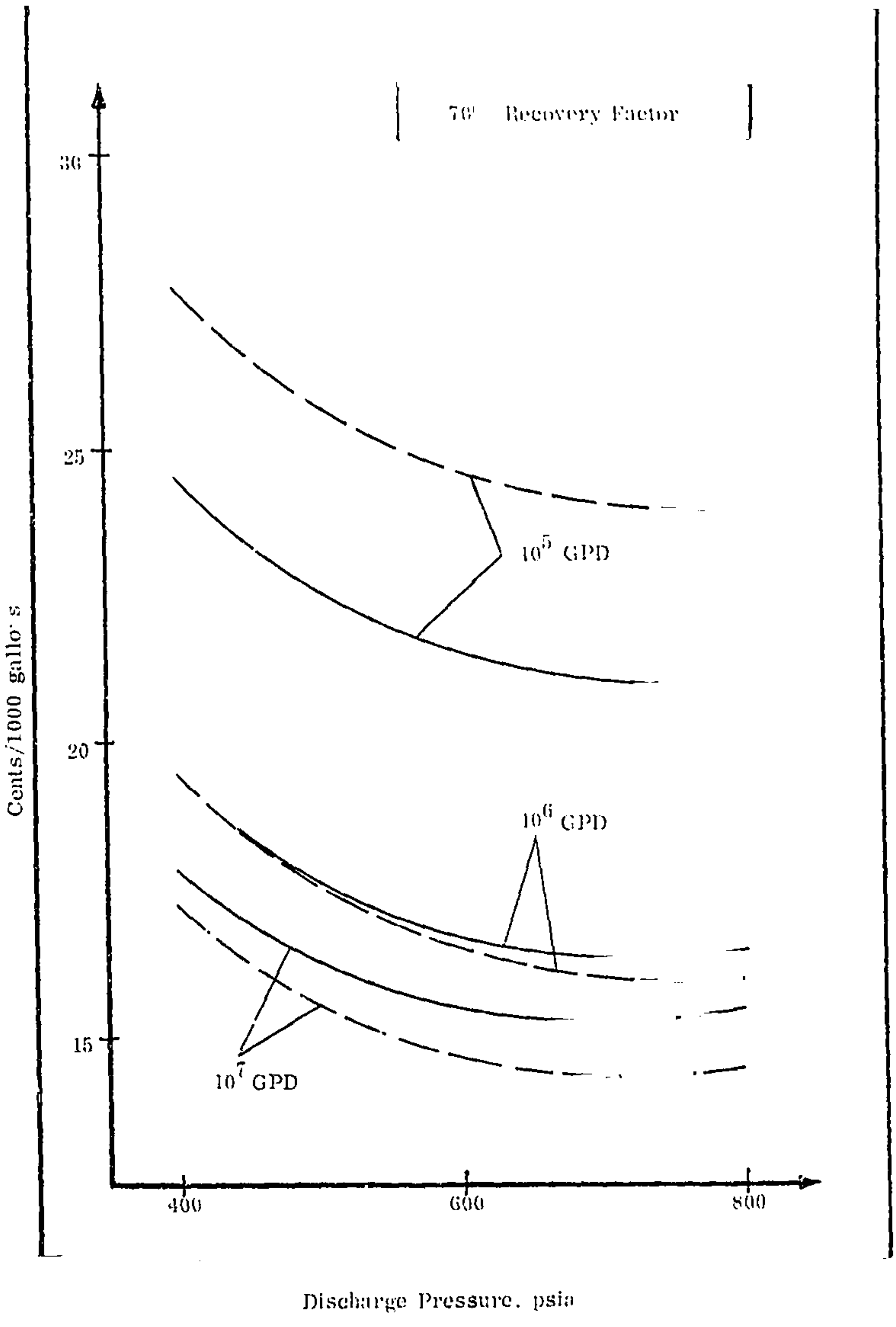


Figure 9.1.9 Contribution to Total Water Cost - Electric Motor Driven Plant with Electric Generator

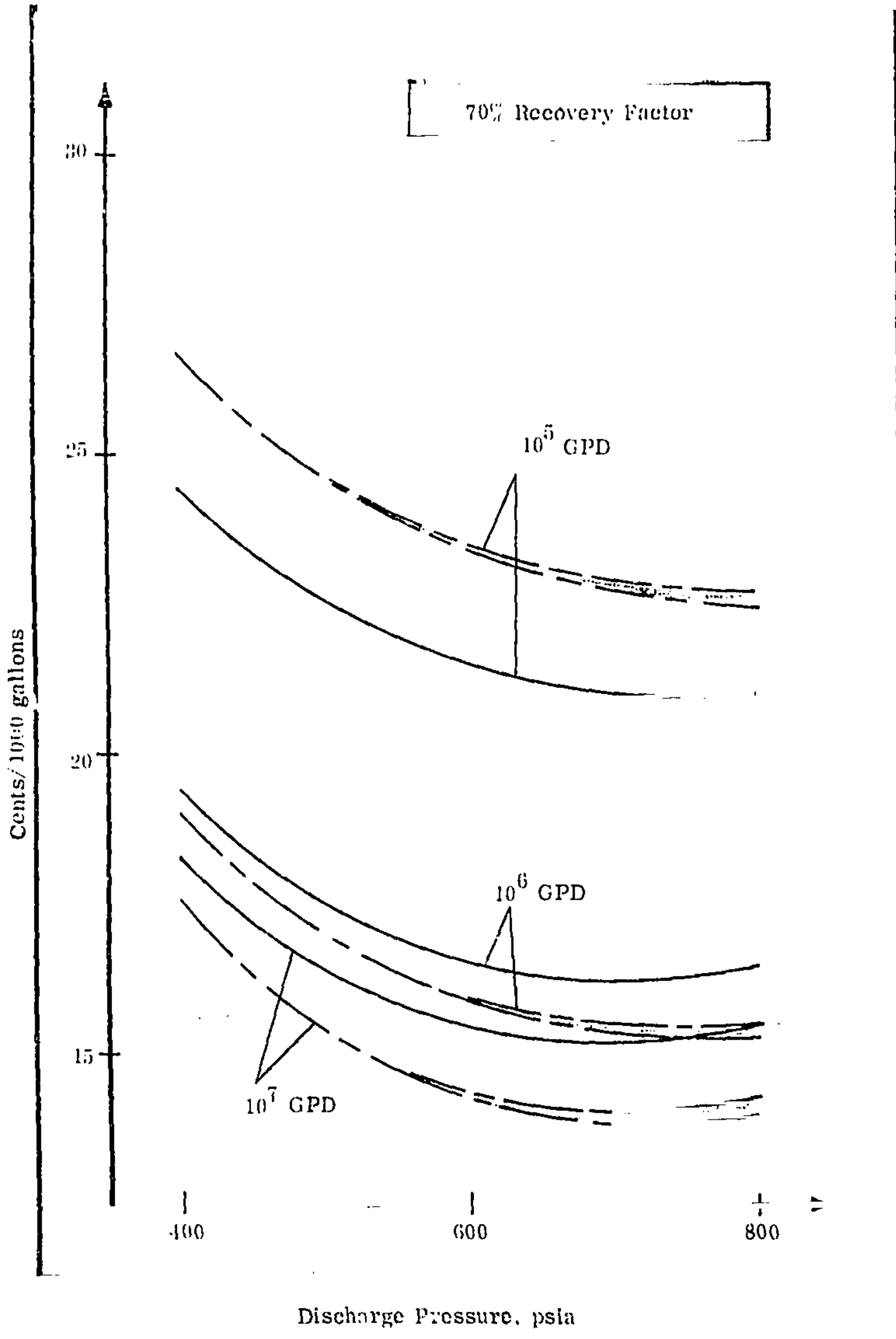


Figure 9. 1. 10 Contribution to Total Water Cost - Electric Motor Driven Plant with Direct Hydraulic Turbine Hook-up

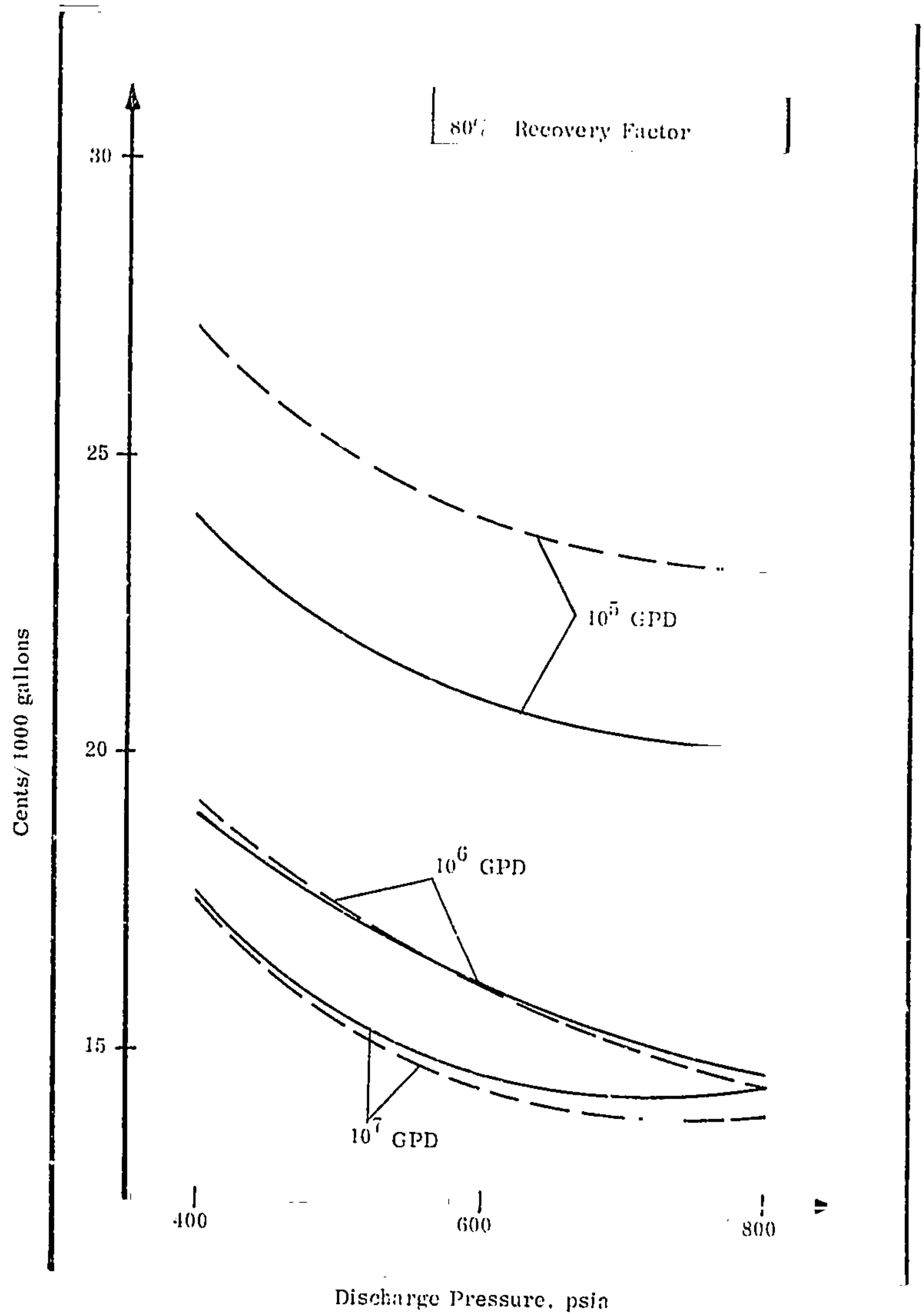


Figure 9.1.11 Contribution to Total Water Cost - Electric Motor Driven Plant with Electric Generator

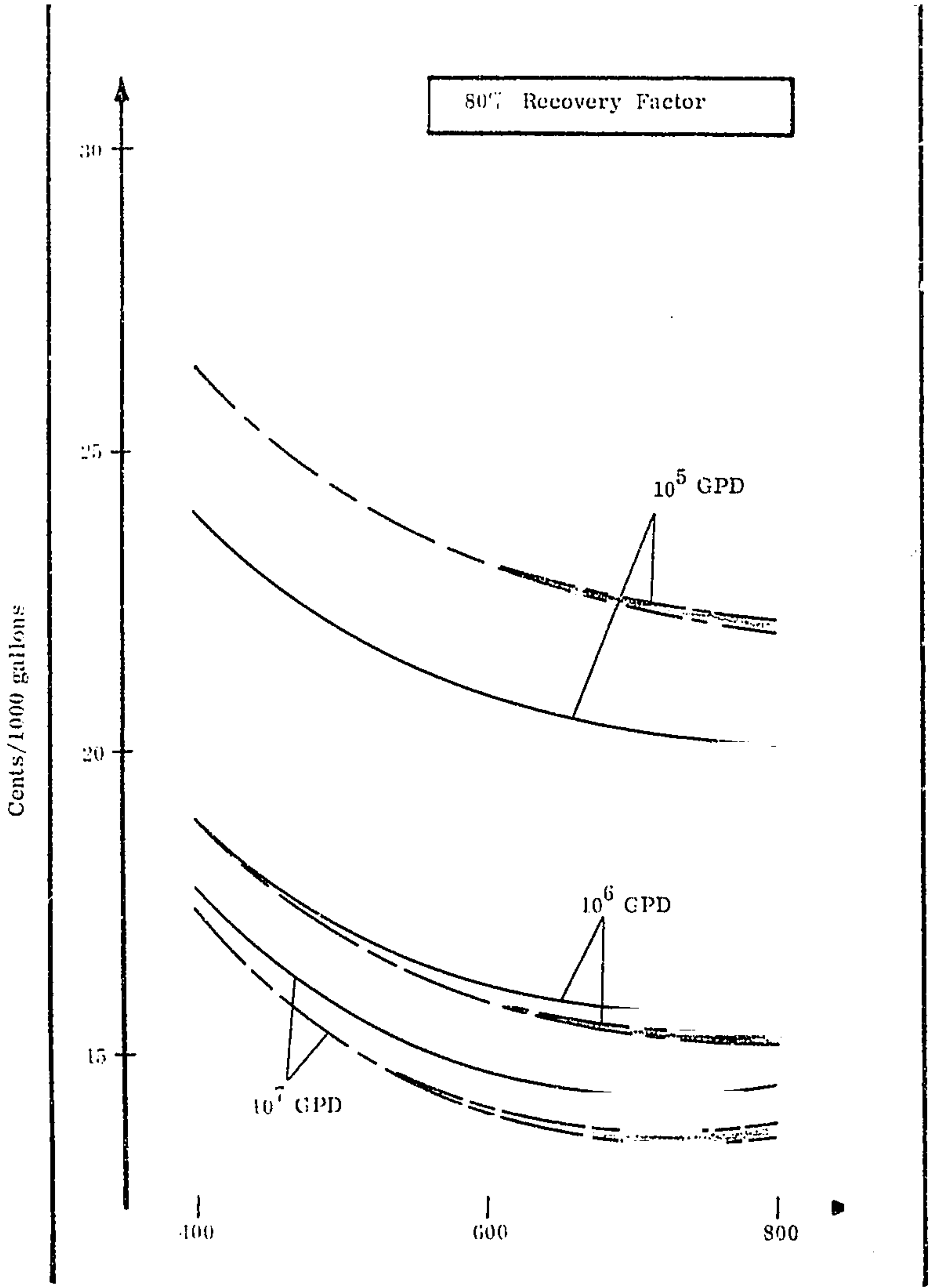


Figure 9.1.12 Contribution to Total Water Cost - Electric Motor Driven Plant with Direct Hydraulic Turbine Hook-up

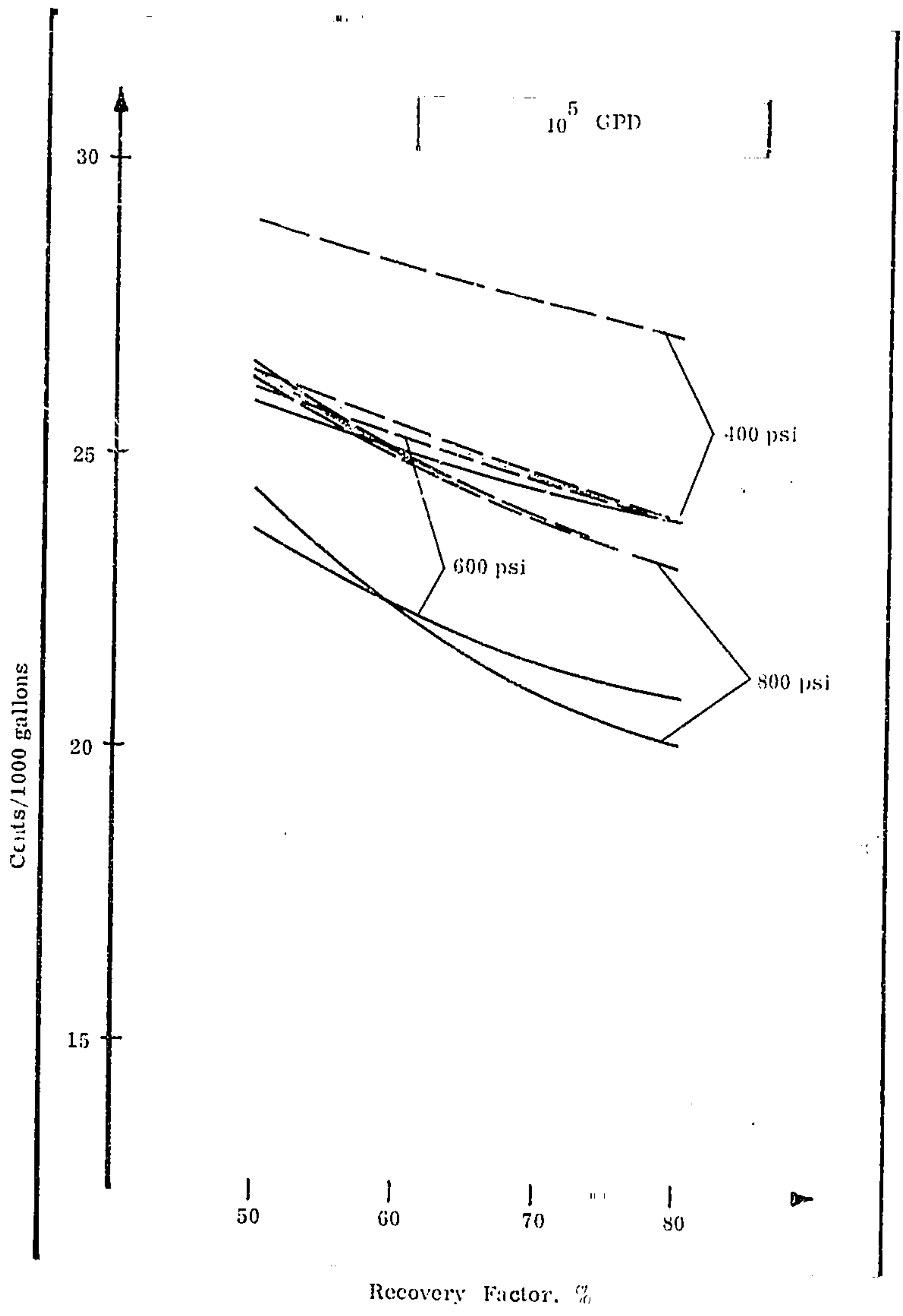


Figure 9.1.13 Contribution to Total Water Cost - Electric Motor Driven Plant with Electric Generator

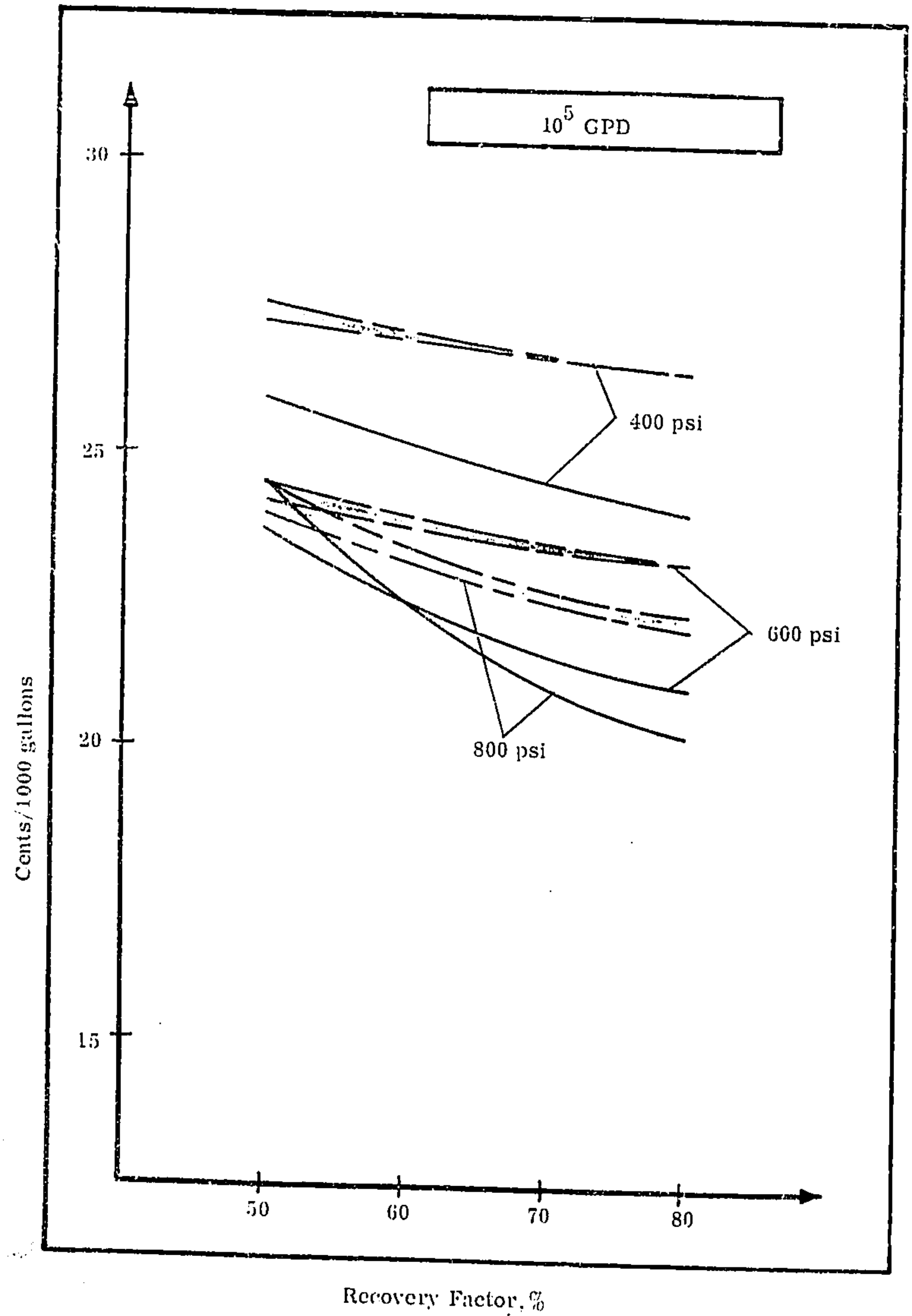


Figure 9. 1. 14 Contribution to Total Water Cost - Electric Motor Driven Plant with Direct Hydraulic Turbine Hook-up

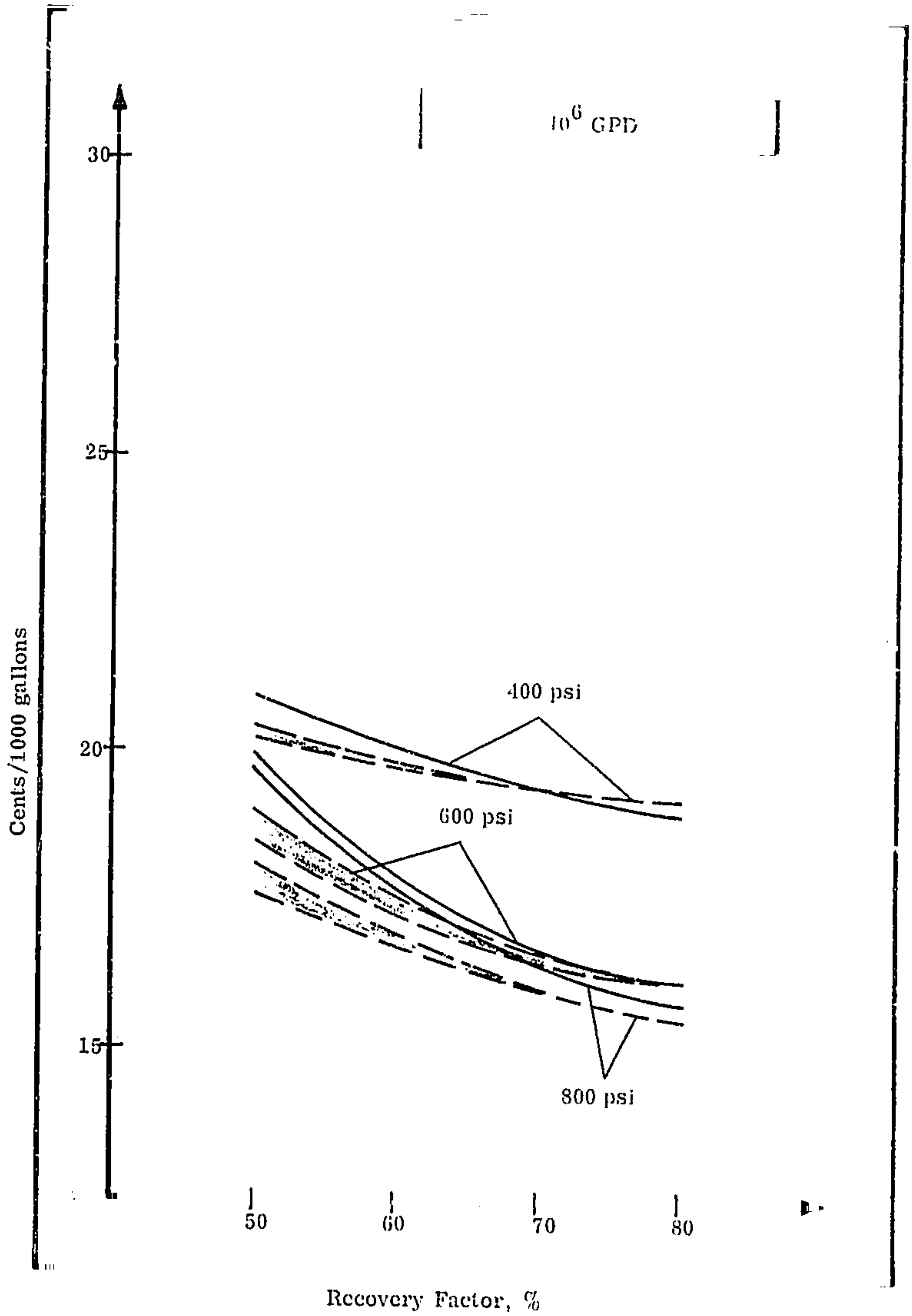


Figure 9. 1. 15 Contribution to Total Water Cost - Electric Motor Driven Plant with Electric Generator

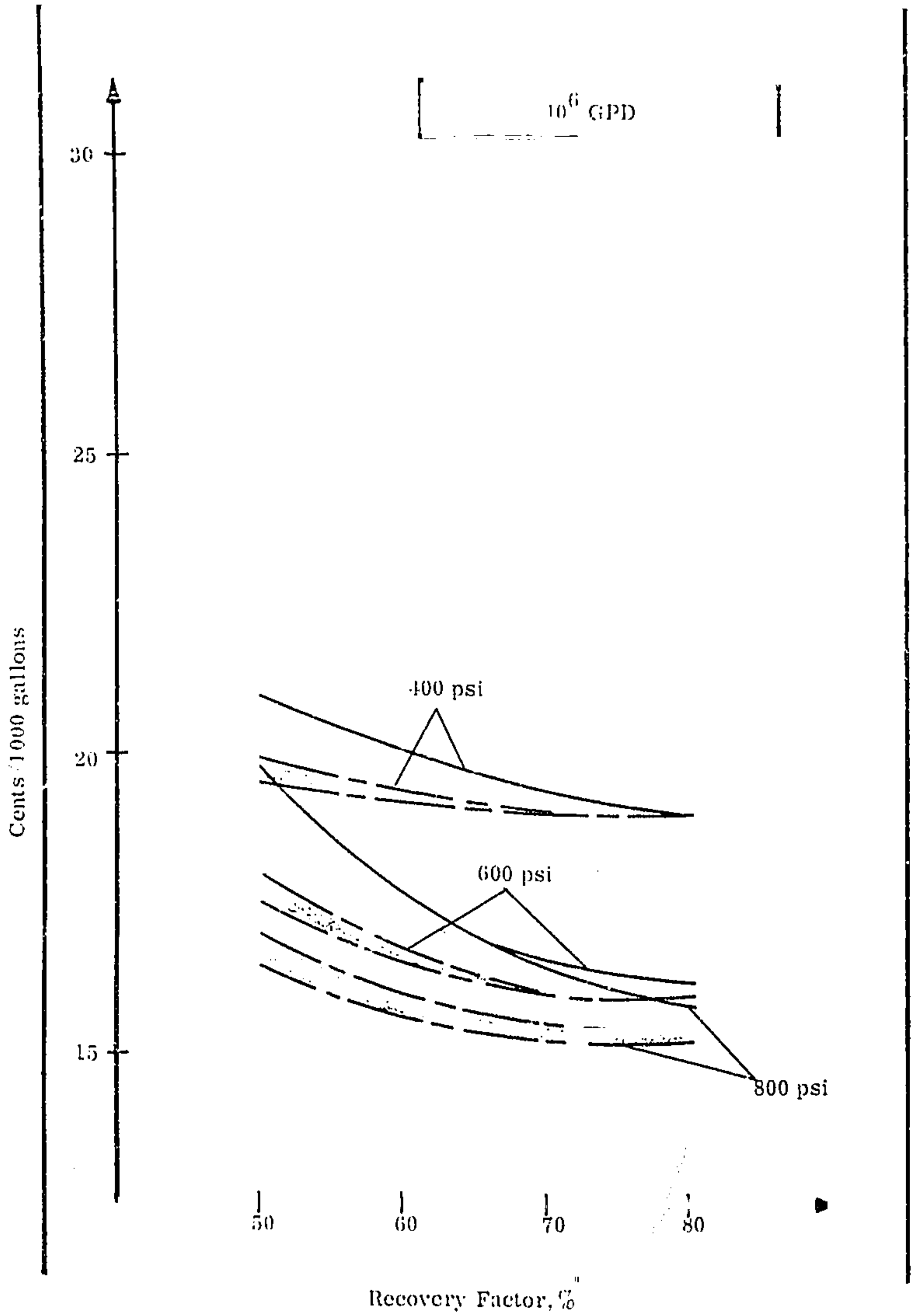


Figure 9.1.16 Contribution to Total Water Cost - Electric Motor Driven Plant with Direct Hydraulic Turbine Hook-up

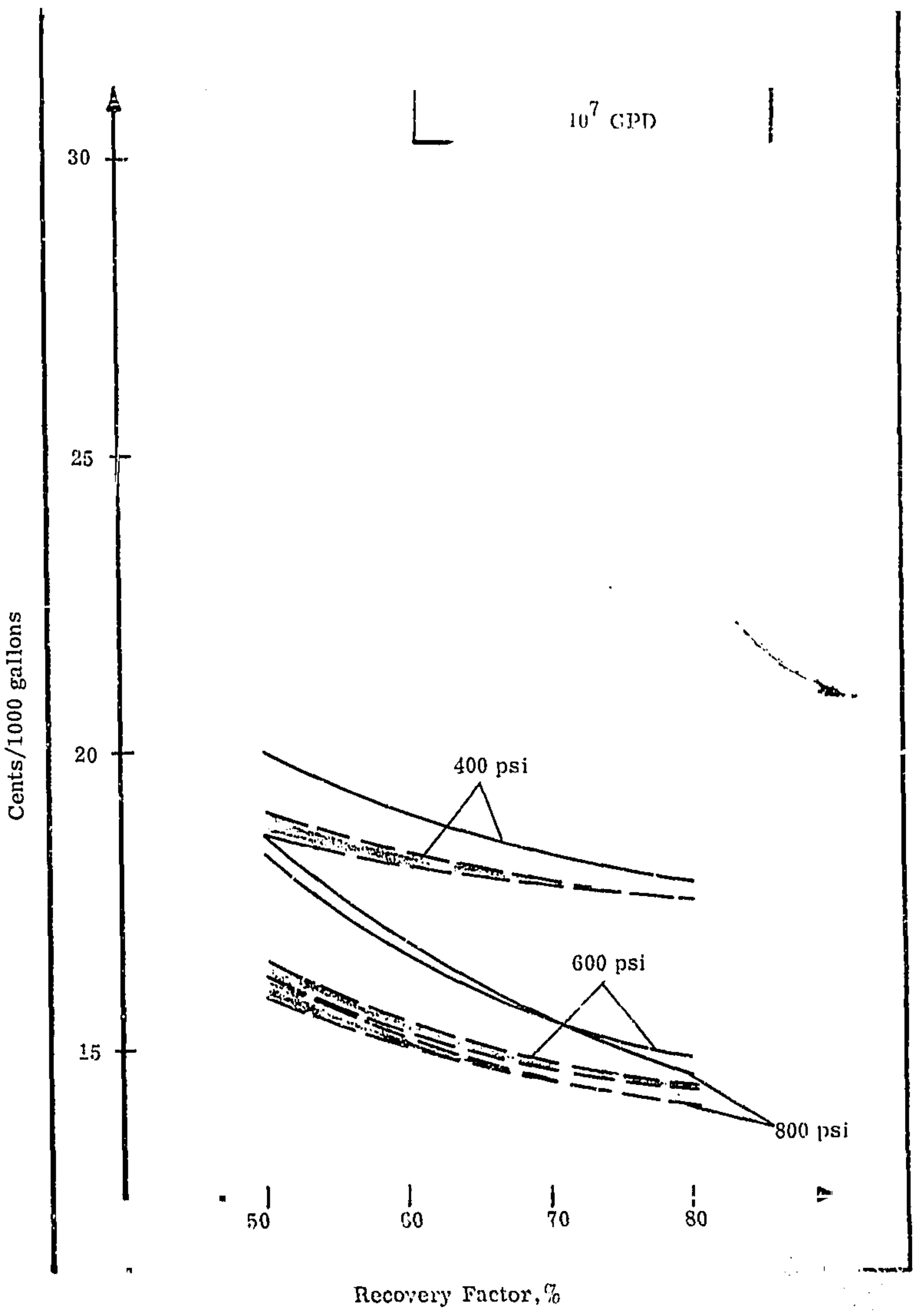


Figure 9. 1. 17 Contribution to Total Water Cost - Electric Motor Driven Plant with Electric Generator

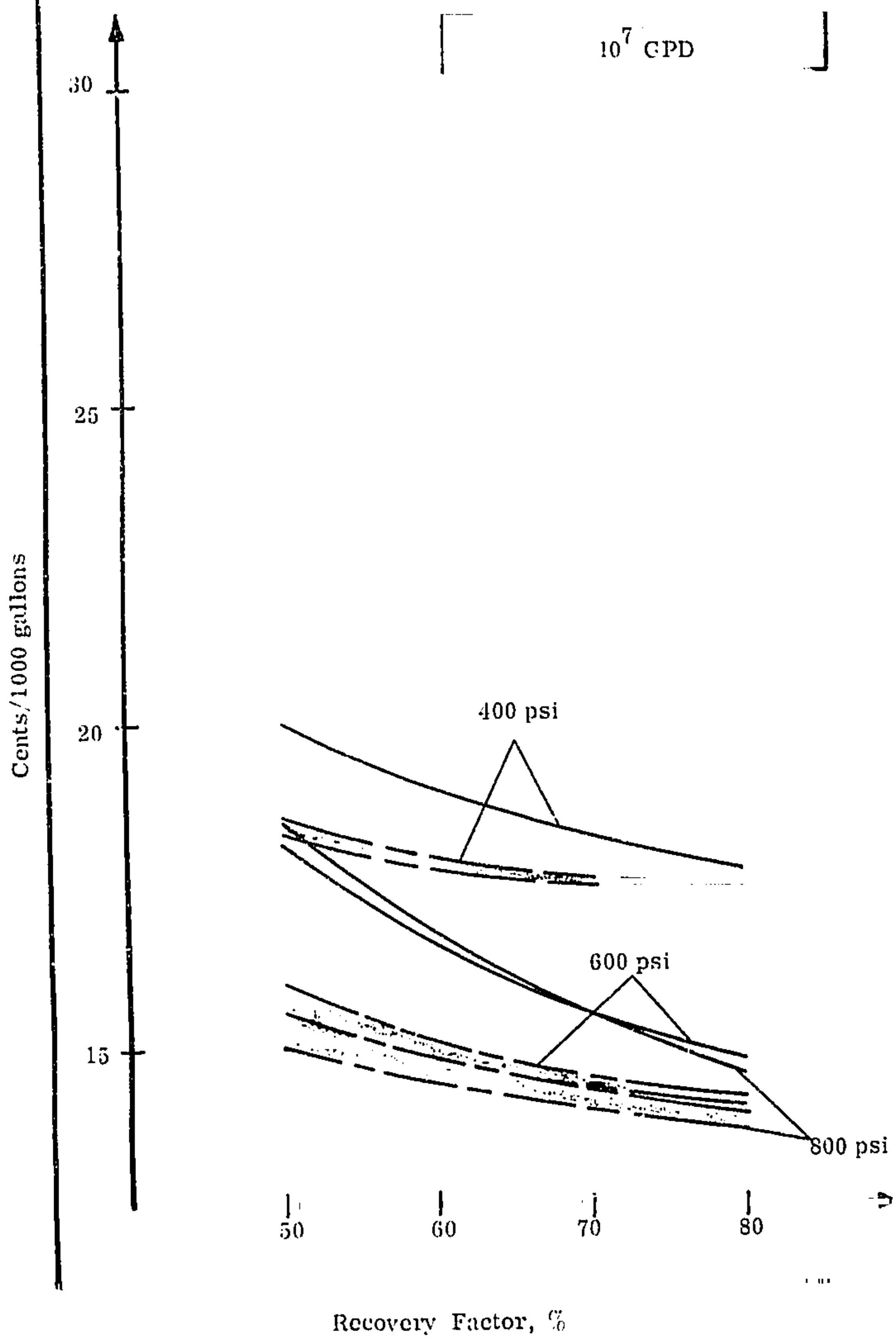


Figure 9. 1. 18 Contribution to Total W·ter Cost - Electric Motor Driven Plant with Direct Hydraulic Turbine Hook-up

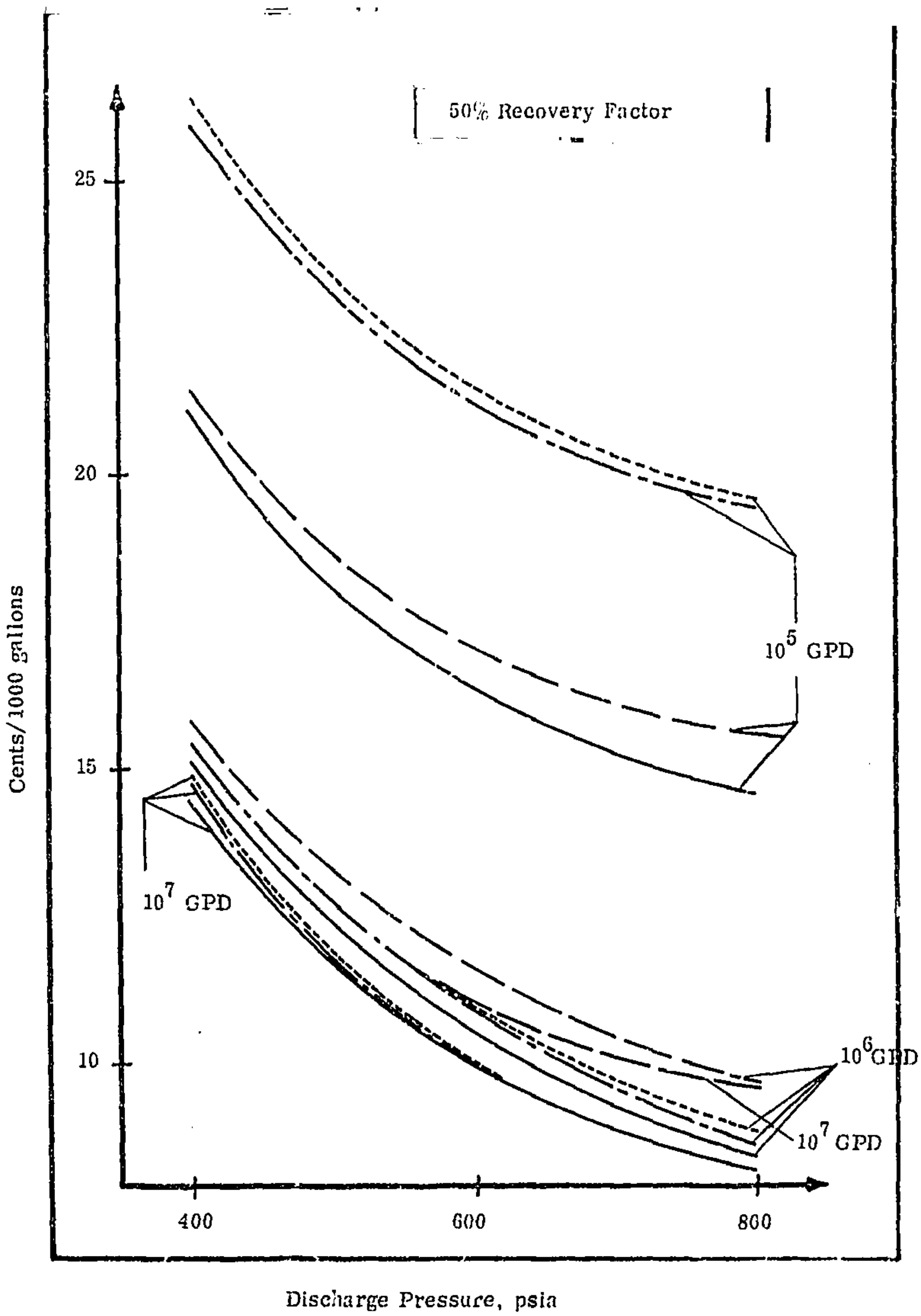


Figure 9.1.19 Contribution to Total Water Cost Exclusive of Power Costs for Various Drivers

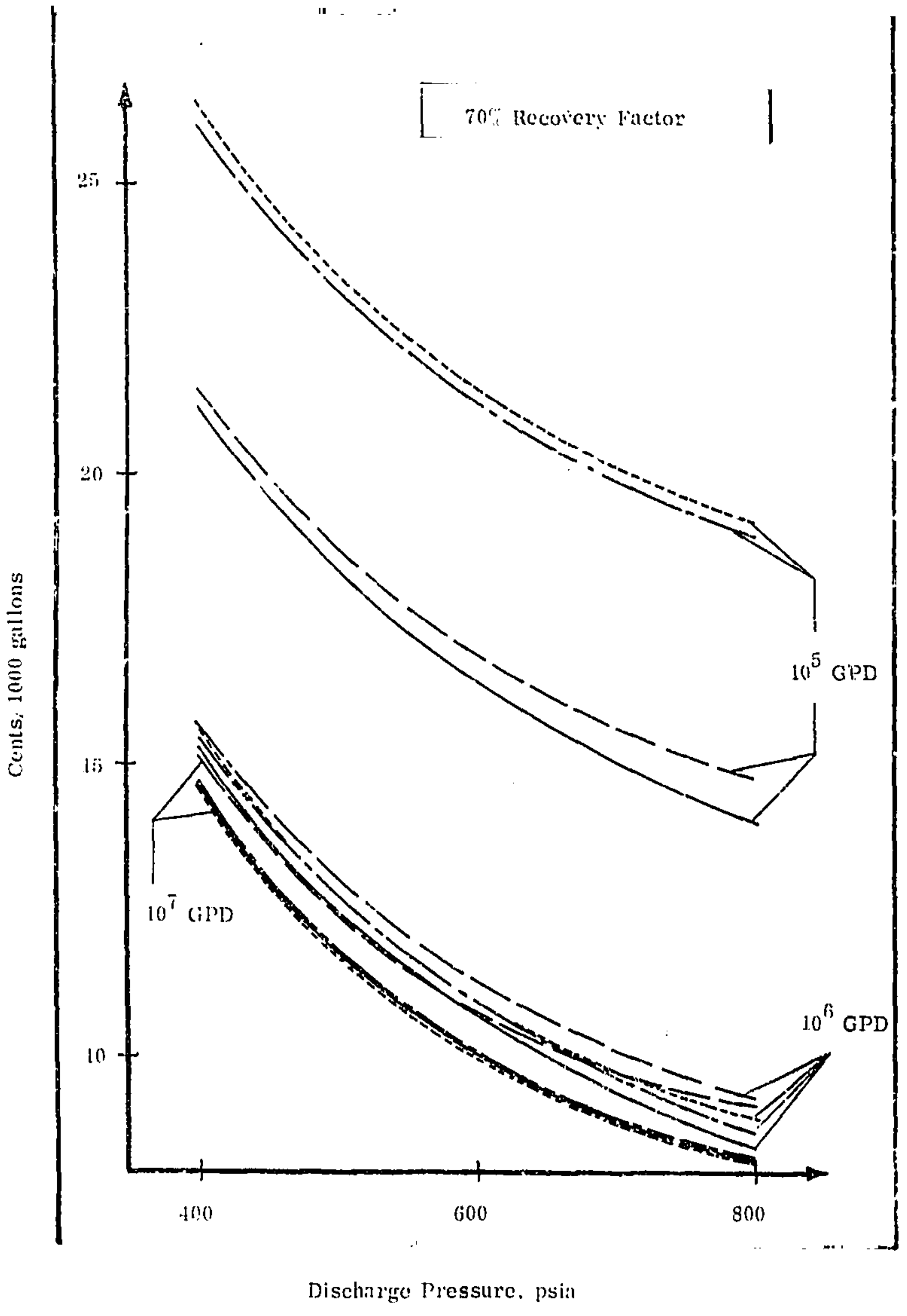


Figure 9.1.20 Contribution to Total Water Cost Exclusive of Power Costs for Various Drivers

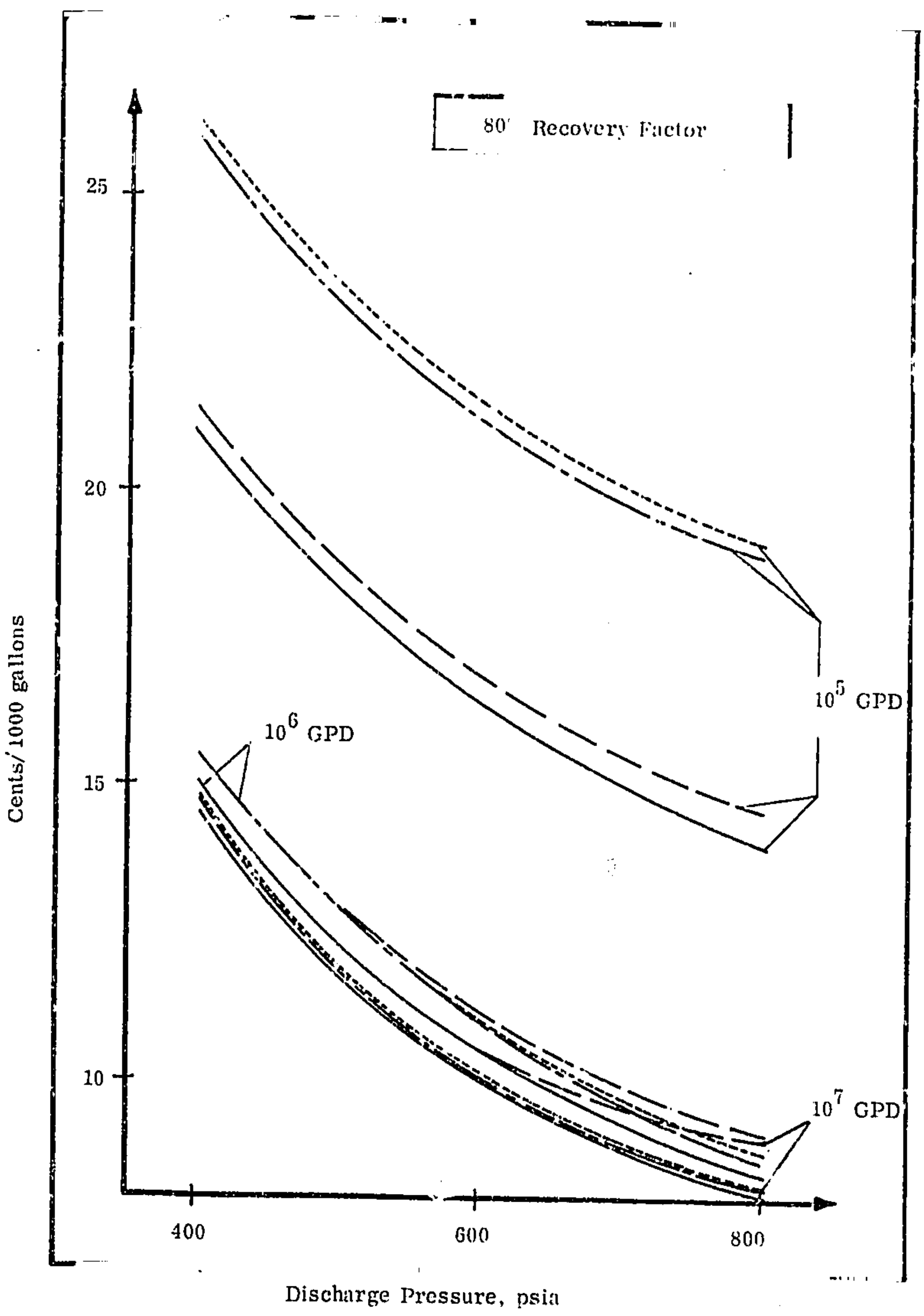


Figure 9.1.21 Contribution to Total Water Cost Exclusive of Power Costs for Various Drivers

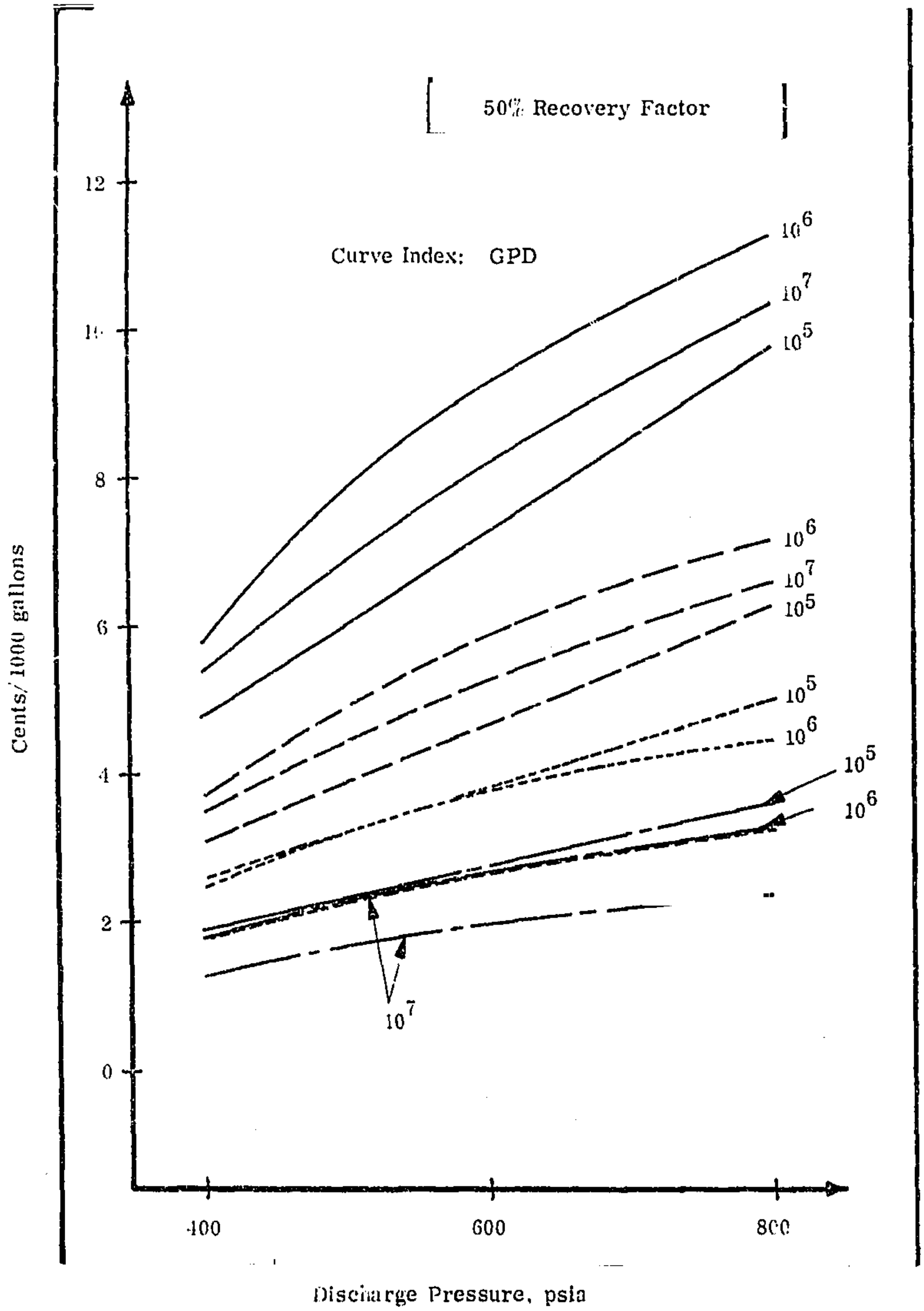


Figure 9.1.22 Power Cost Contribution to Total Water Cost
No Energy Recovery System

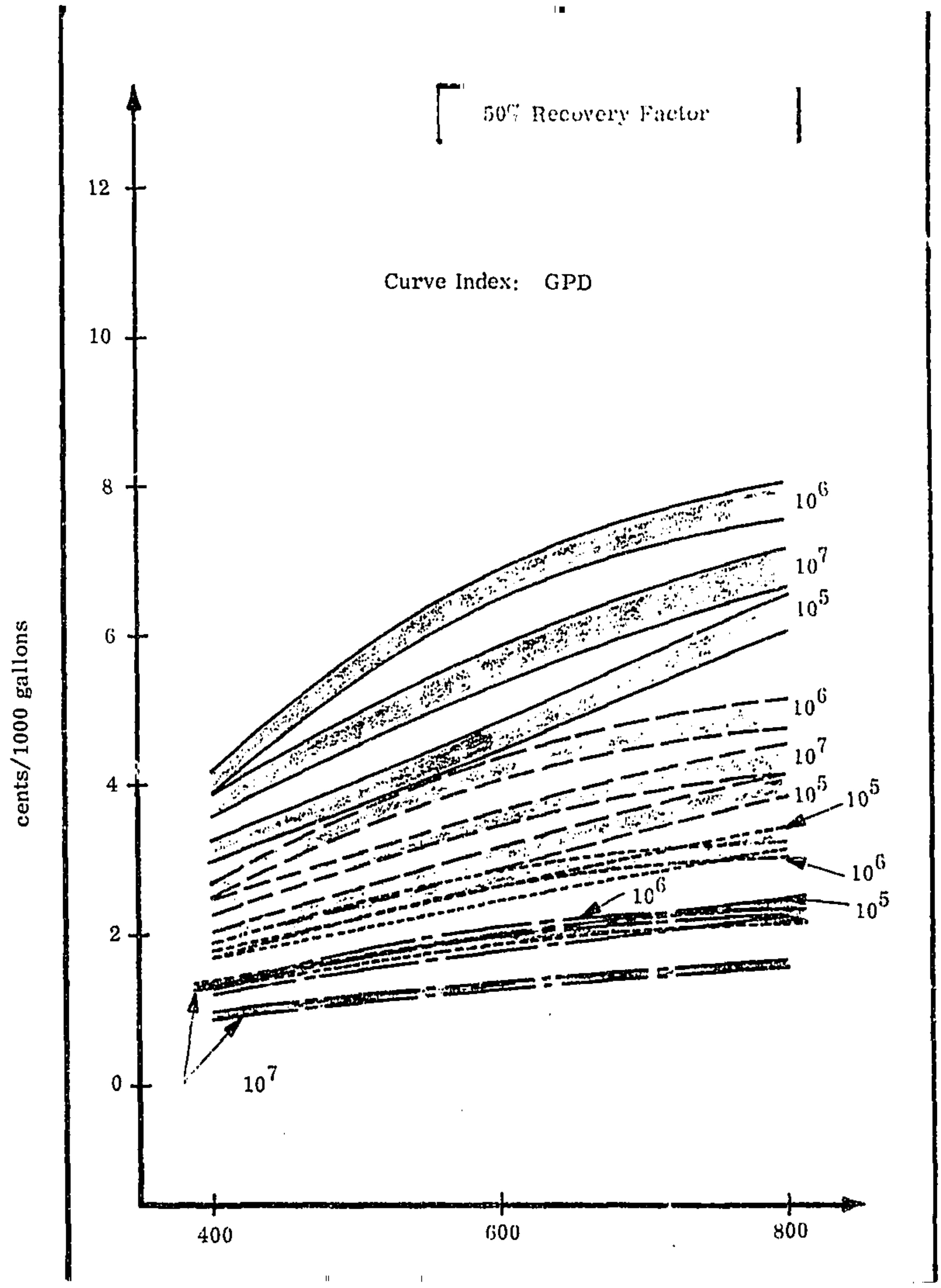
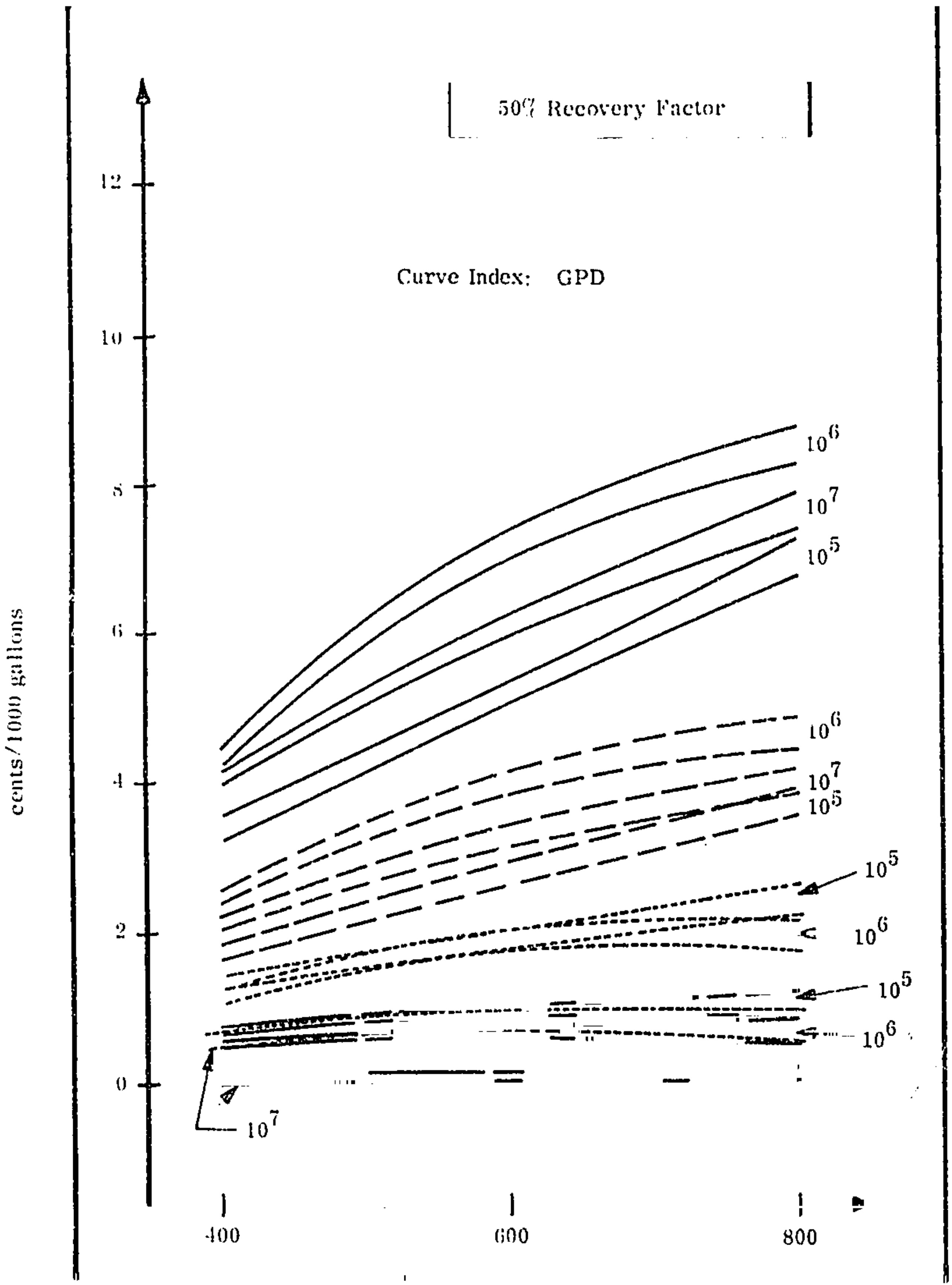


Figure 9.1.23 Power Cost Contribution to Total Water Cost - Energy Recovery System with Direct Hydraulic Hook-up



Discharge Pressure, psia

Figure 9.1.24 Power Cost Contribution to Total Water Cost - Energy Recovery System with Electric Generator

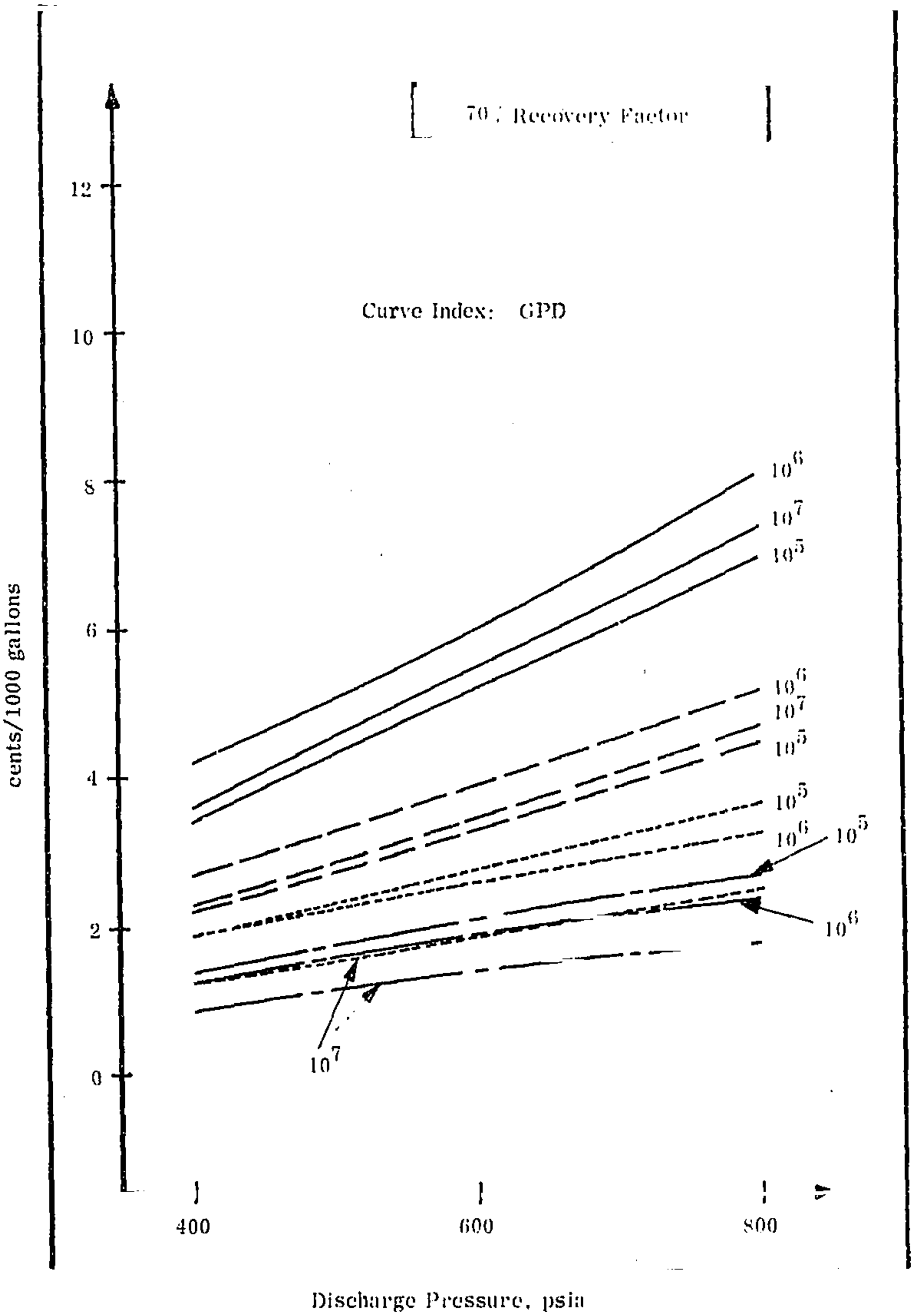


Figure 9.1.25 Power Cost Contribution to Total Water Cost - No Energy Recovery System

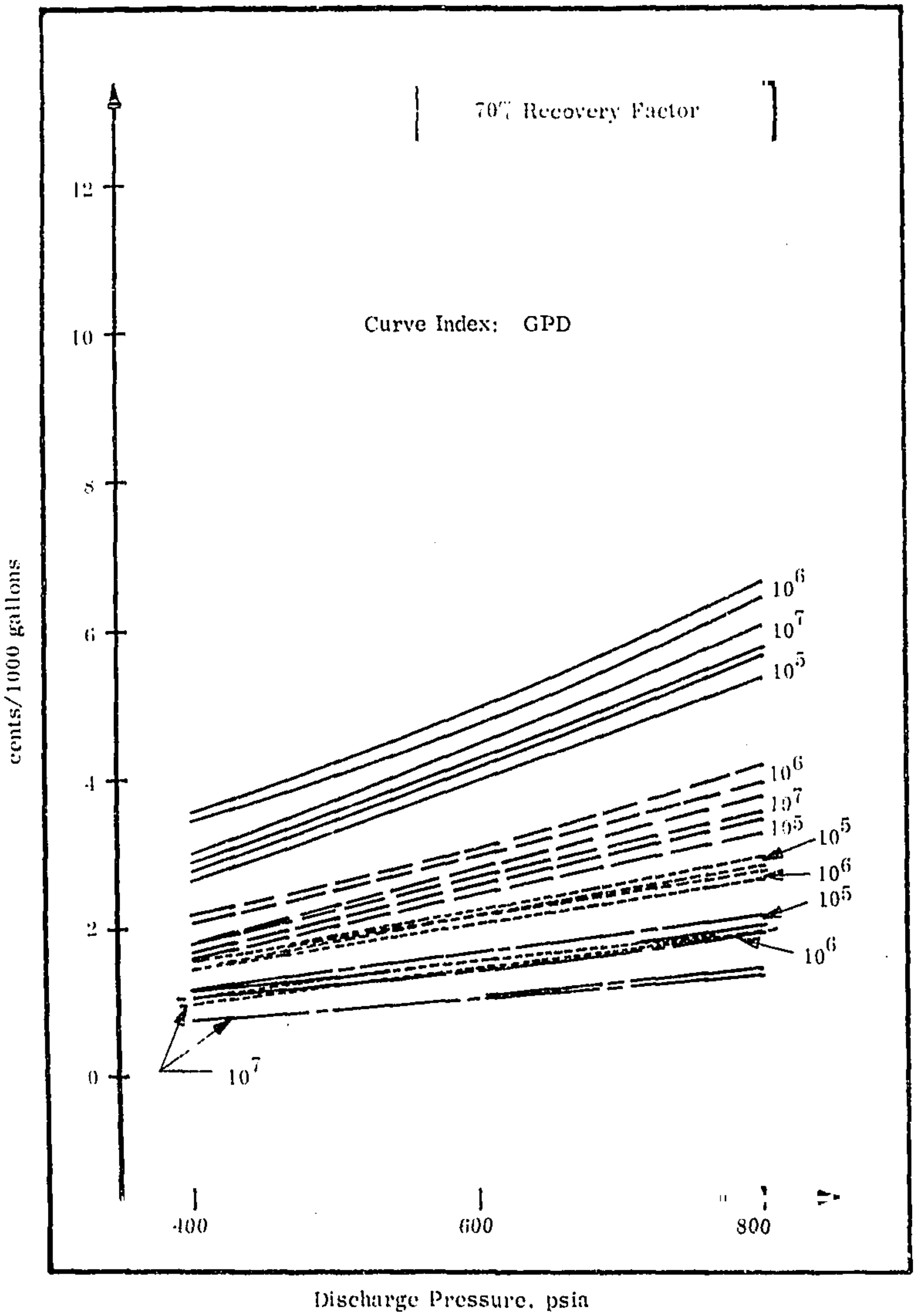
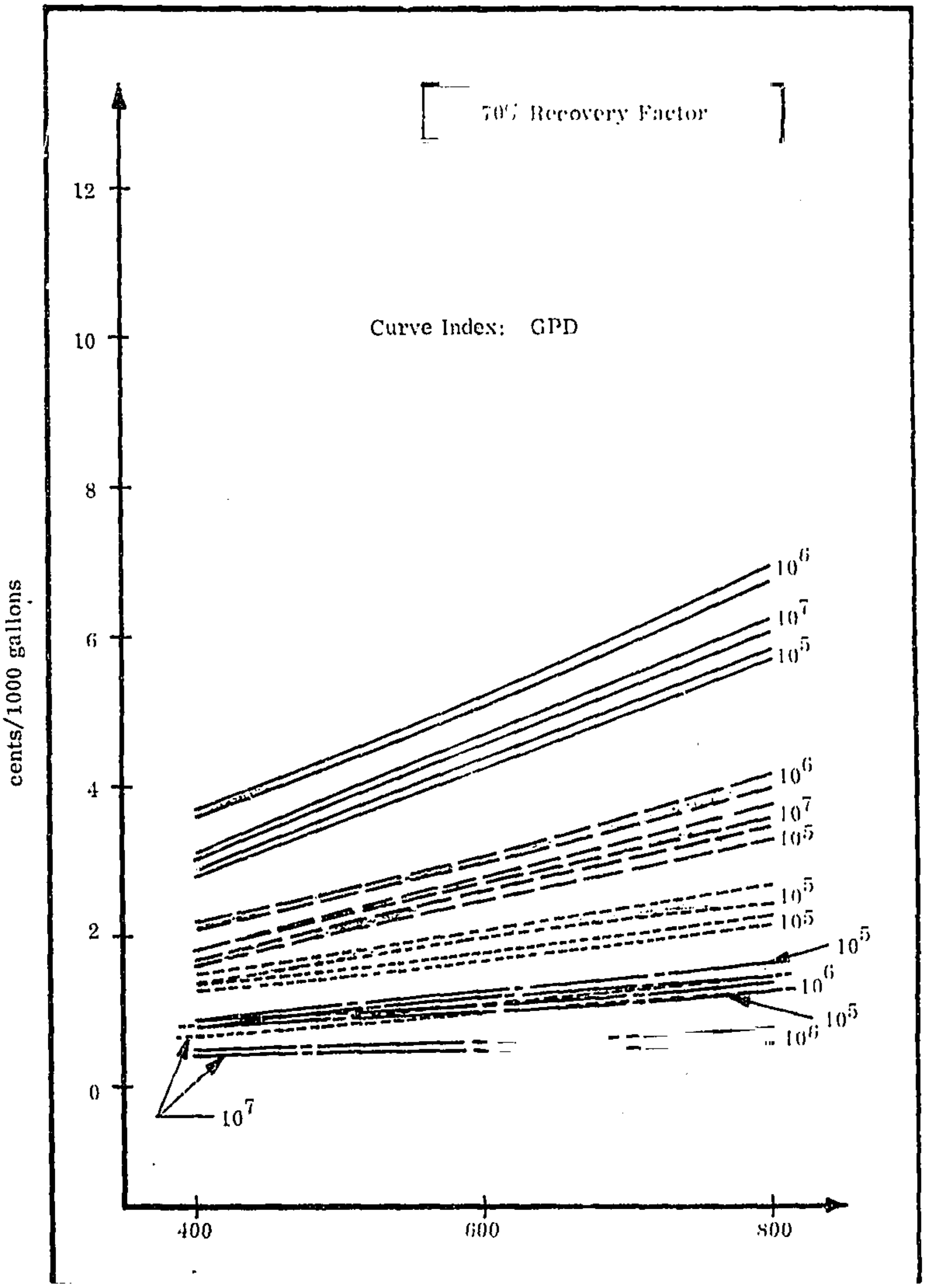
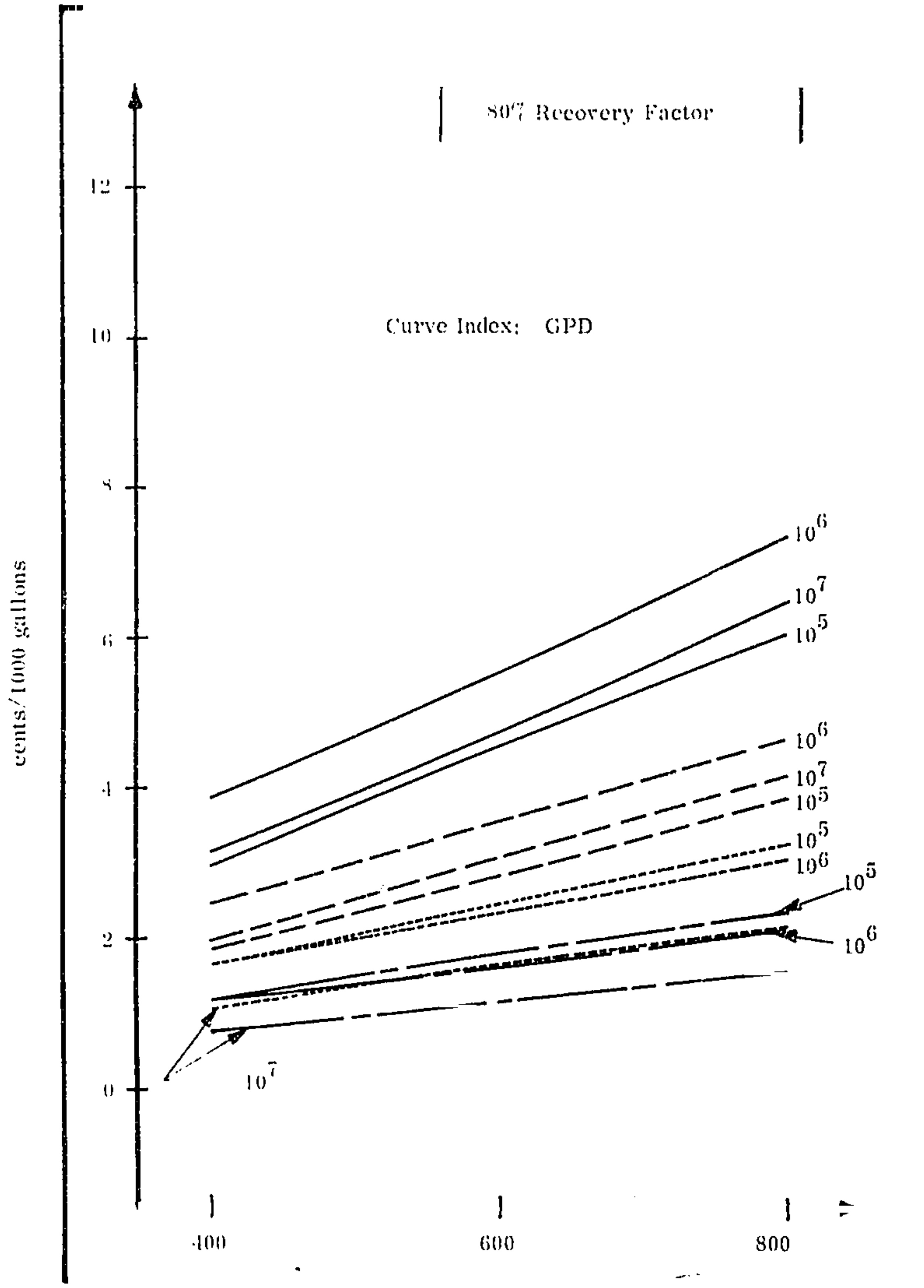


Figure 9.1.26 Power Cost Contribution to Total Water Cost - Energy Recovery System with Direct Hydraulic Hook-up



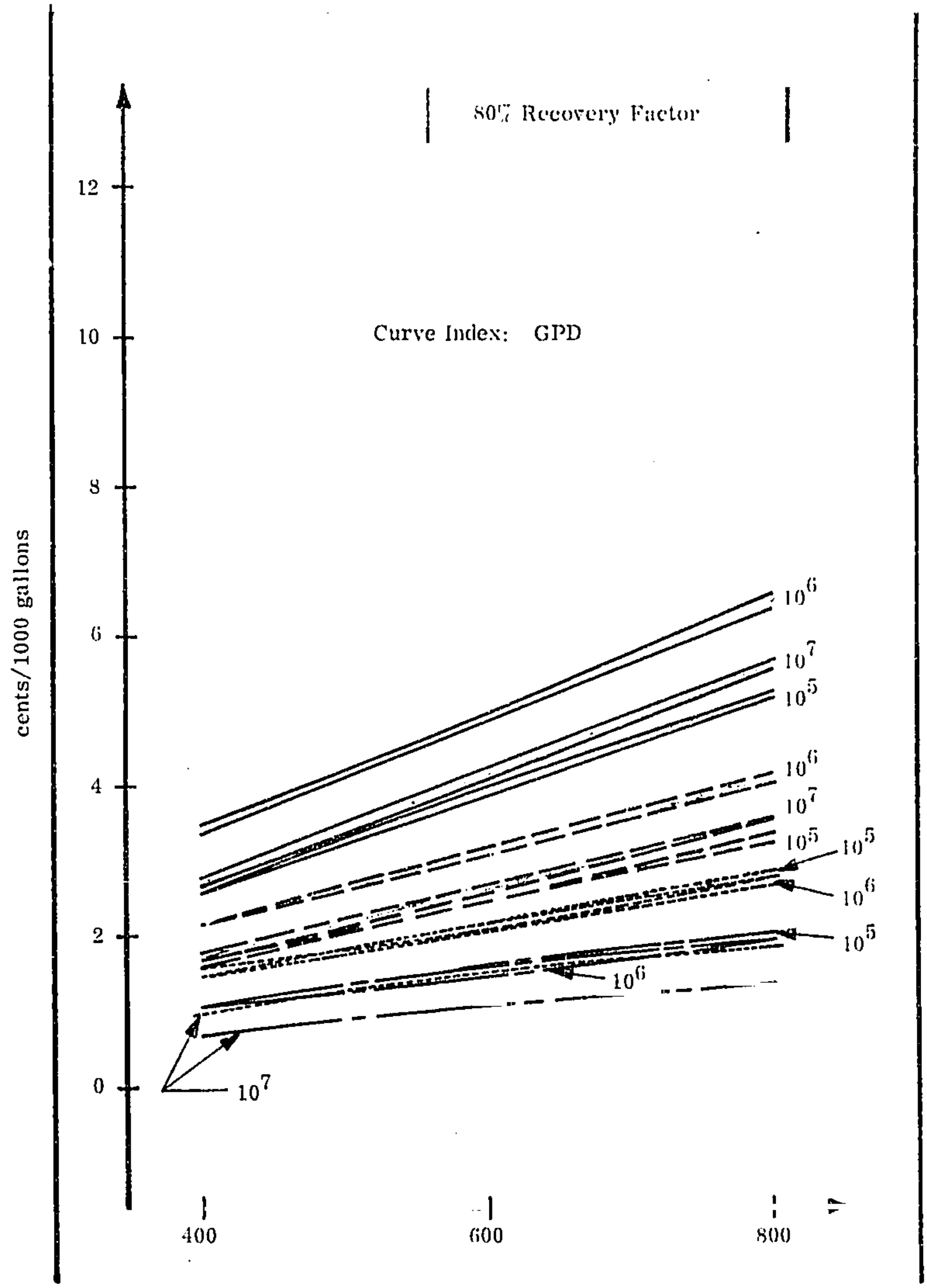
Discharge Pressure, psia

Figure 9.1.27 Power Cost Contribution to Total Water Cost - Energy Recovery System with Electric Generator



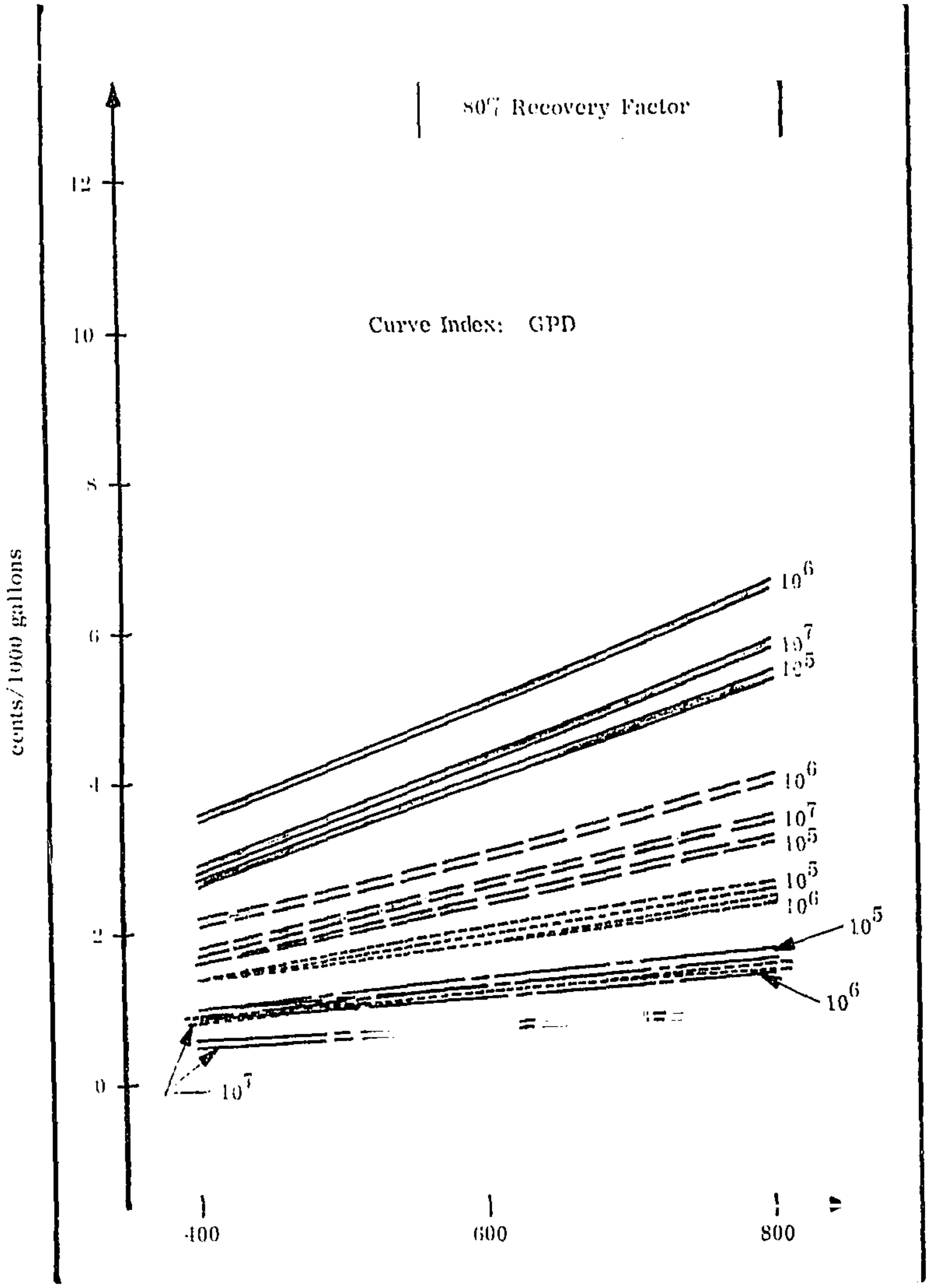
Discharge Pressure, psia

Figure 9.1.28 Power Cost Contribution to Total Water-Energy Recovery System



Discharge Pressure, psia

Figure 9.1.29 Power Cost Contribution to Total Water Cost - Energy Recovery System with Direct Hydraulic Hook-up



Discharge Pressure, psia

Figure 9.1.30 Power Cost Contribution to Total Water Cost - Energy Recovery System with Electric Generator

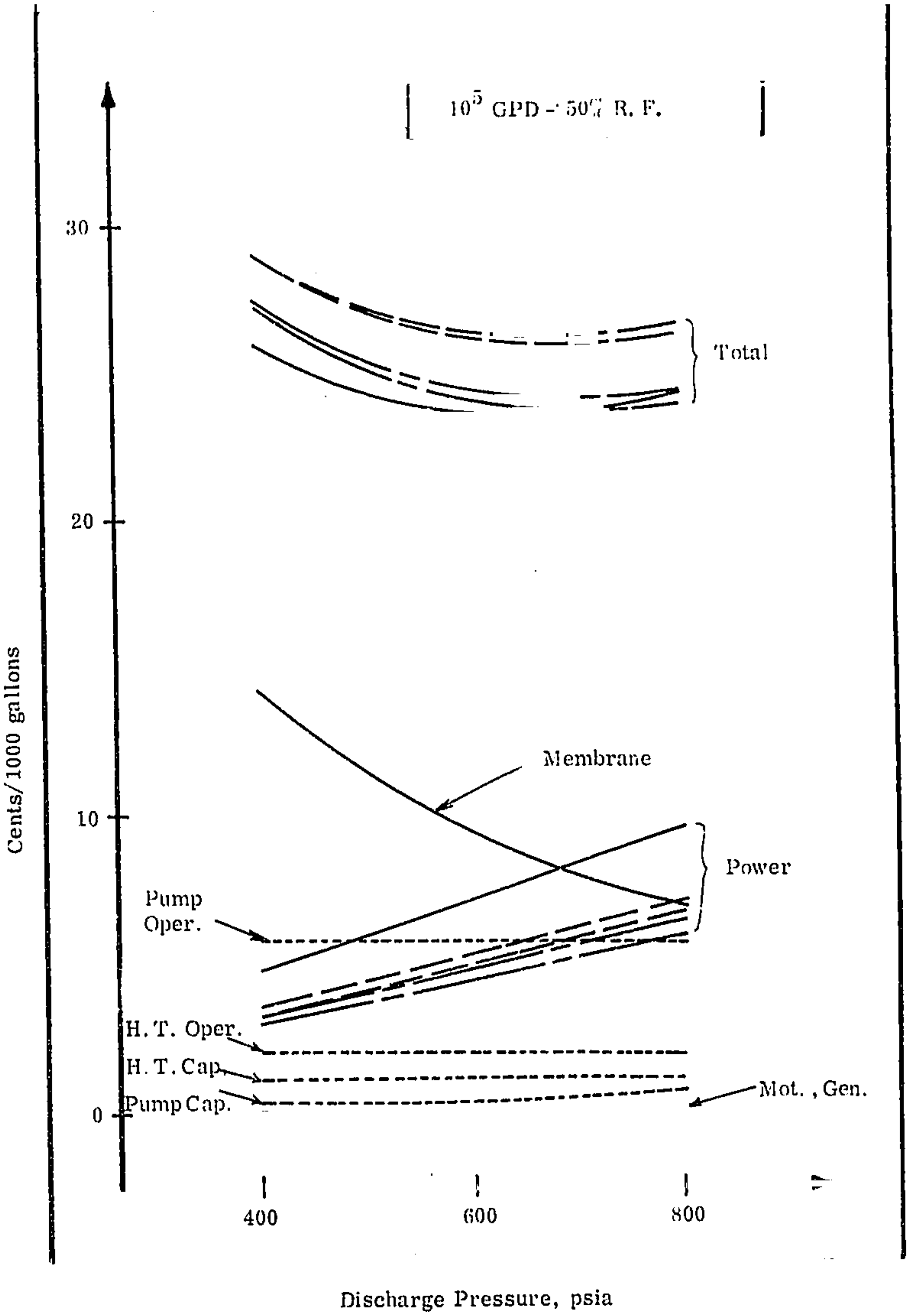
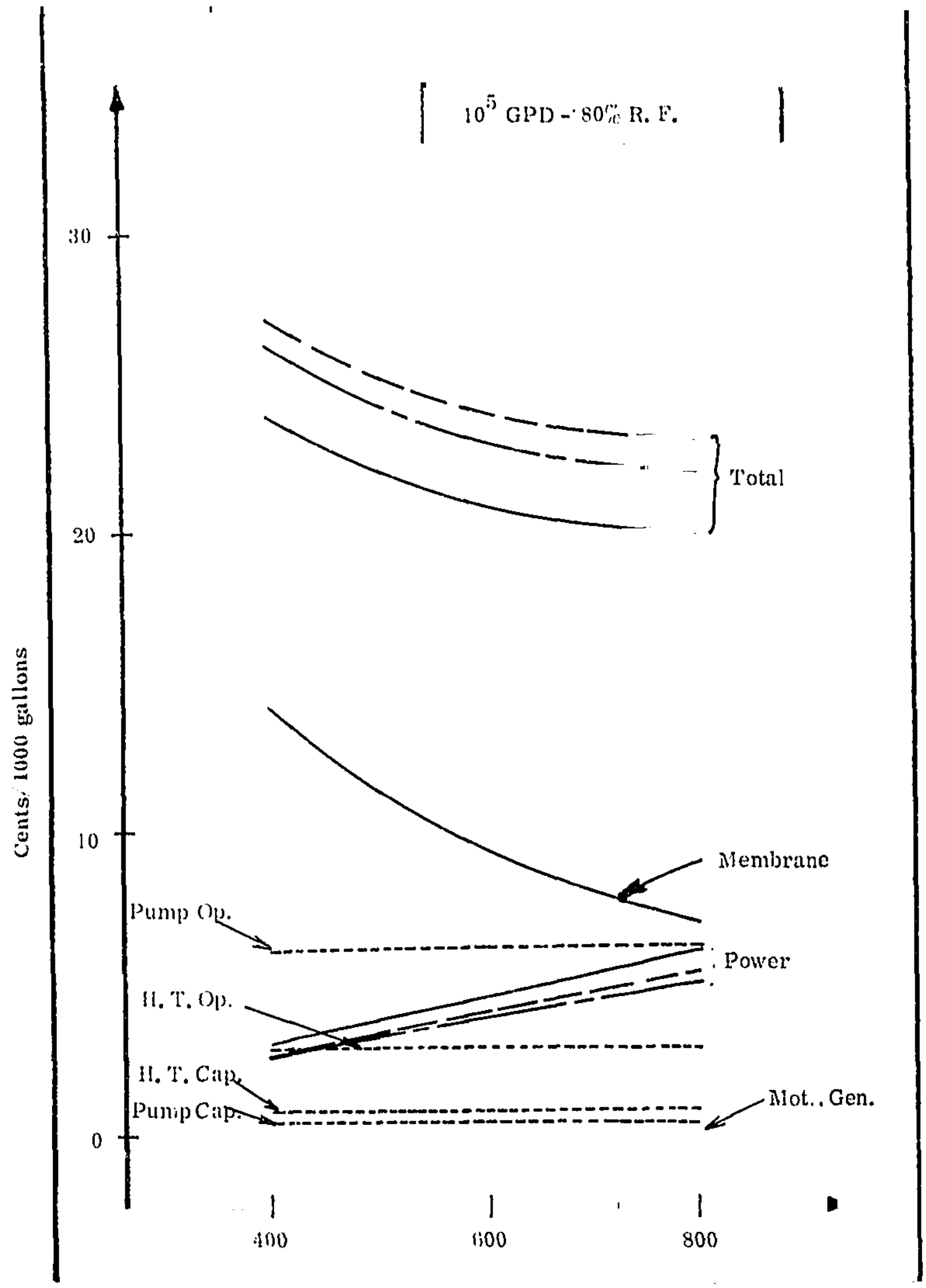


Figure 9.1.31 Contribution to Total Water Cost - Individual Contributions of Capital, Operating and Power Costs Electric Motor Drive



Discharge Pressure, psia

Figure 9.1.32 Contribution to Total Water Cost - Individual Contributions of Capital, Operating and Power Costs Electric Motor Drive

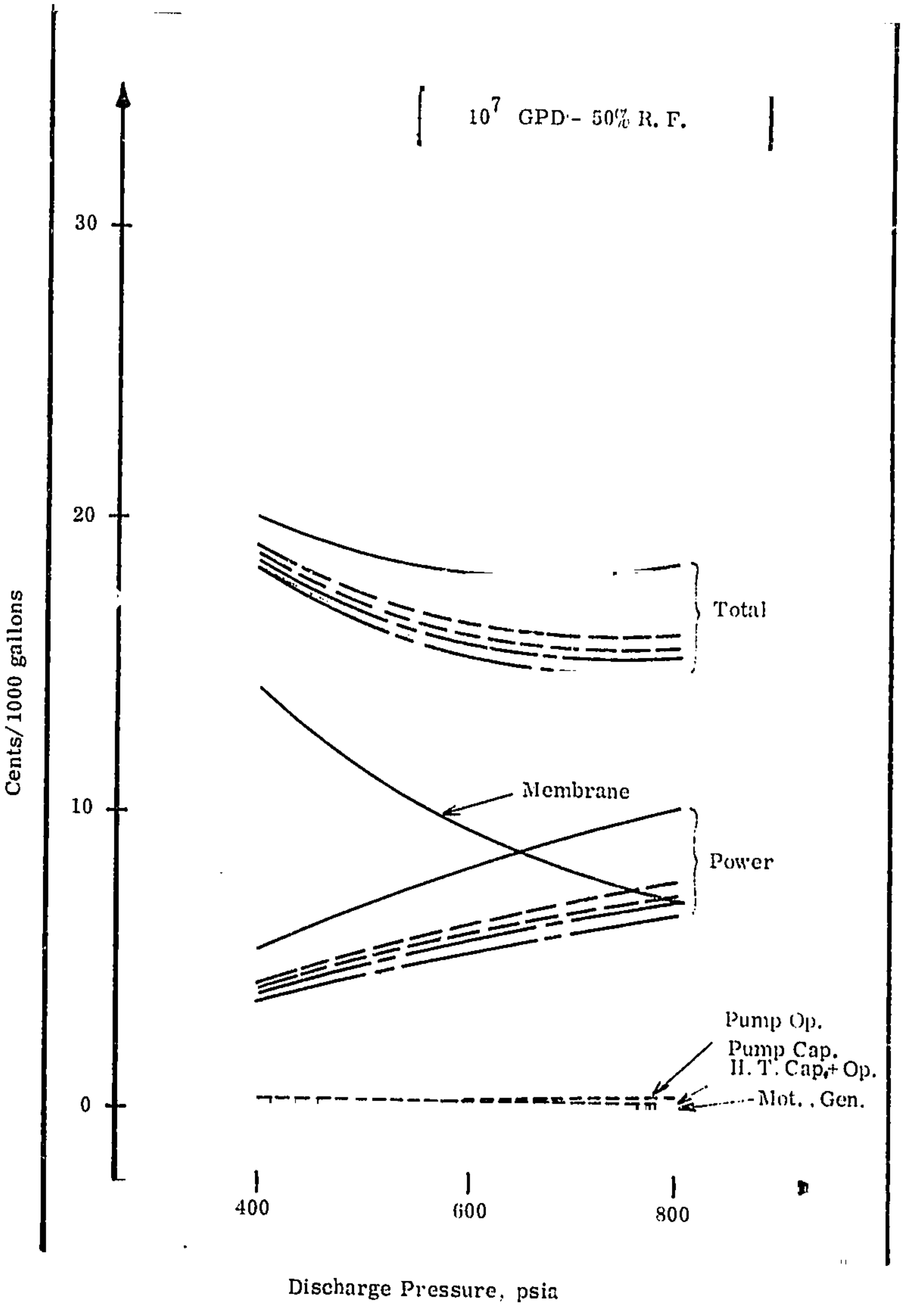
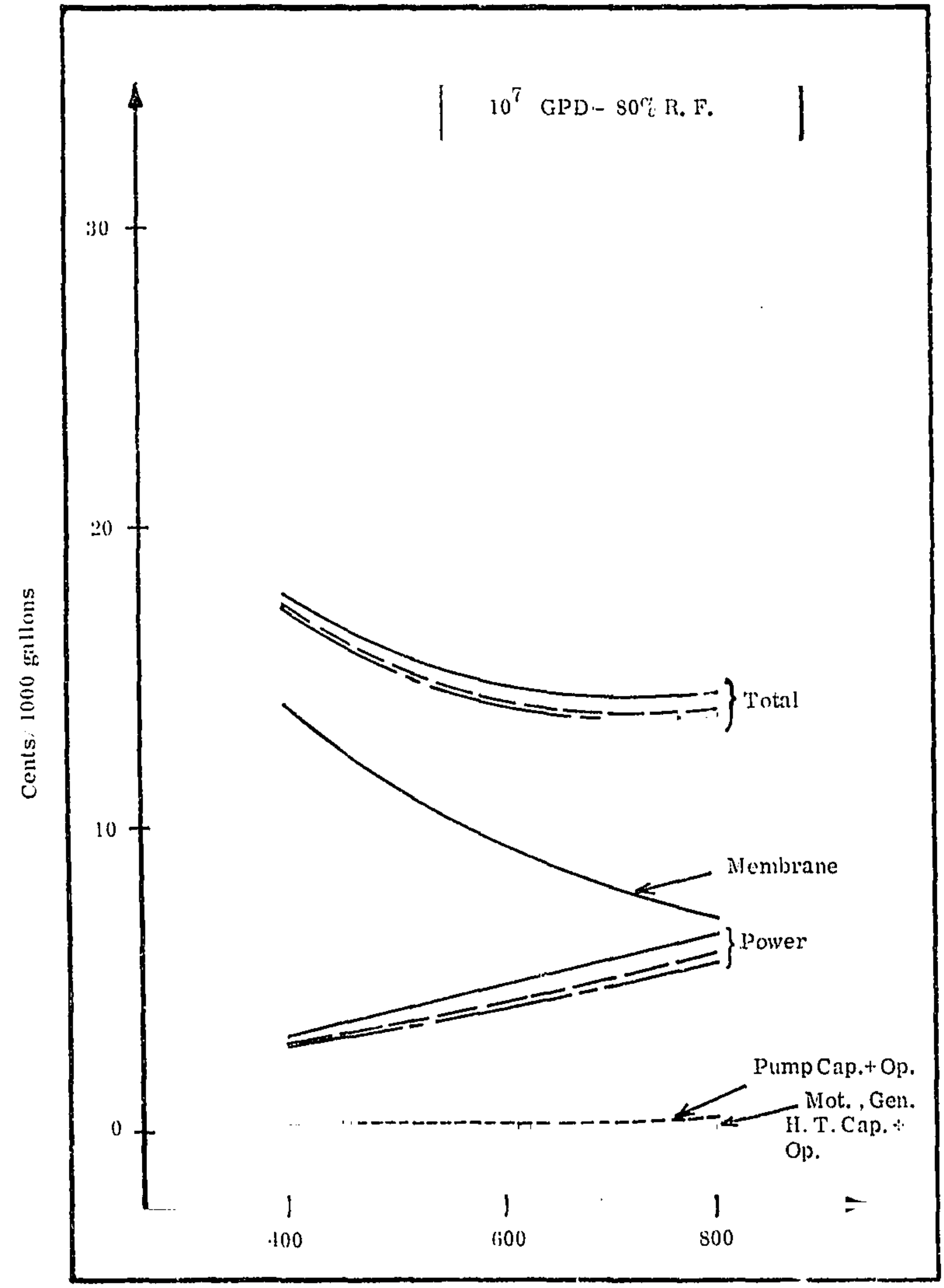


Figure 9.1.33 Contribution to Total Water Cost - Individual Contributions of Capital, Operating and Power Costs Electric Motor Drive



Discharge Pressure, psia

Figure 9.1.34 Contribution to Total Water Cost - Individual Contributions of Capital, Operating and Power Costs Electric Motor Drive

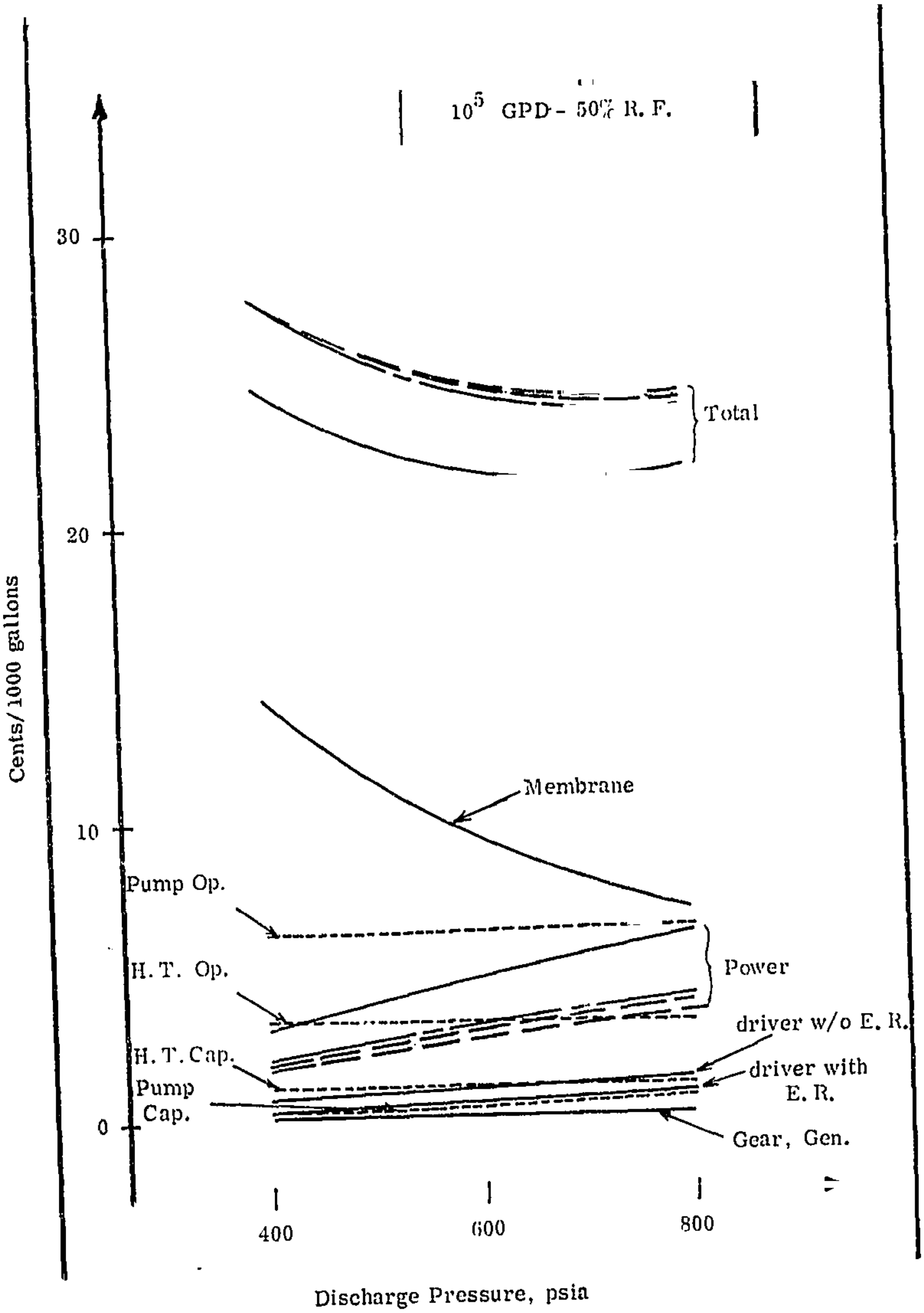
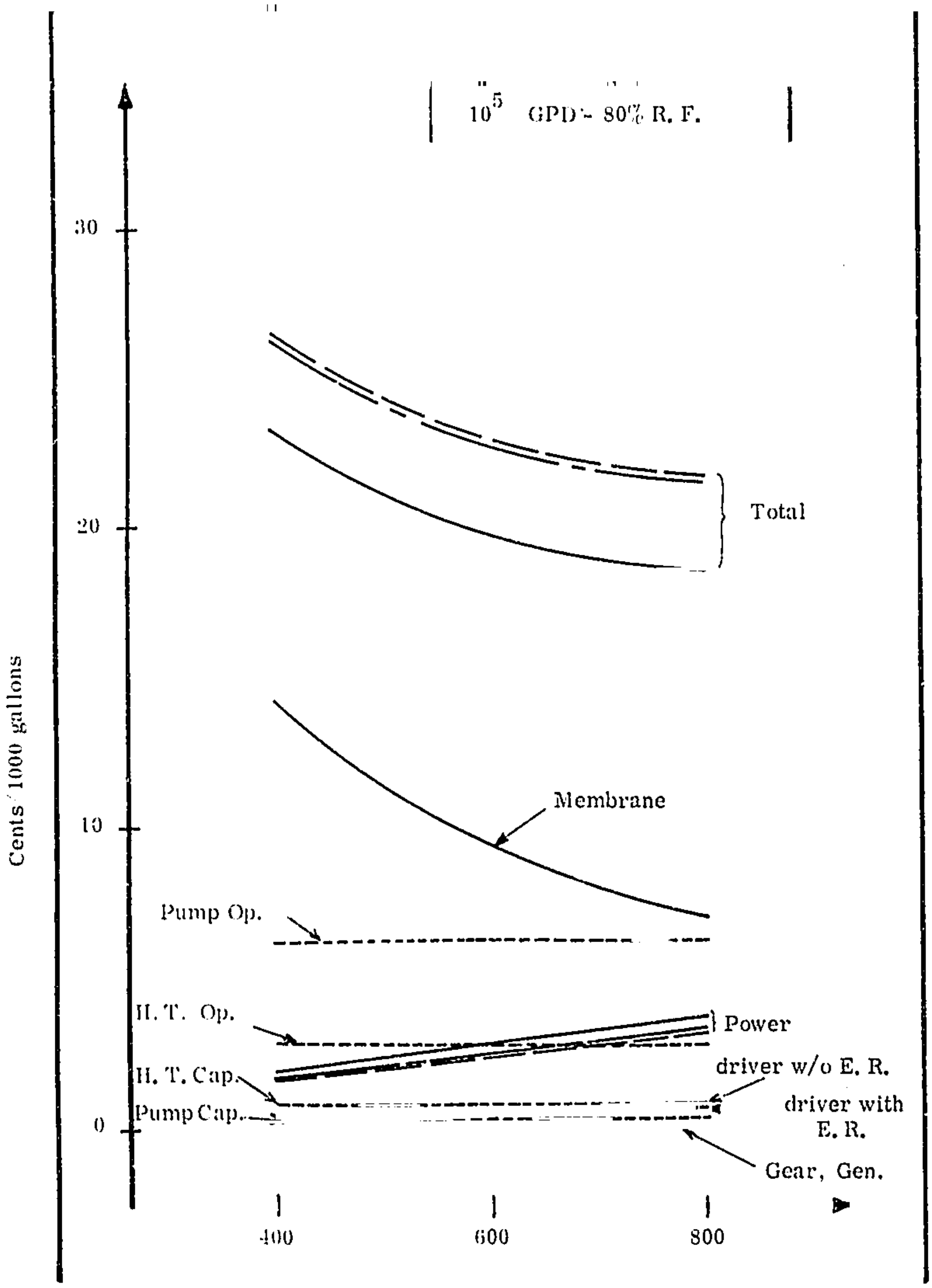


Figure 9.1.35 Contribution to Total Water Cost - Individual Contributions of Capital, Operating and Power Costs Diesel Engine Drive



Discharge Pressure, psia

Figure 9.1.36 Contribution to Total Water Cost - Individual Contributions of Capital, Operating and Power Costs Diesel Engine Drive

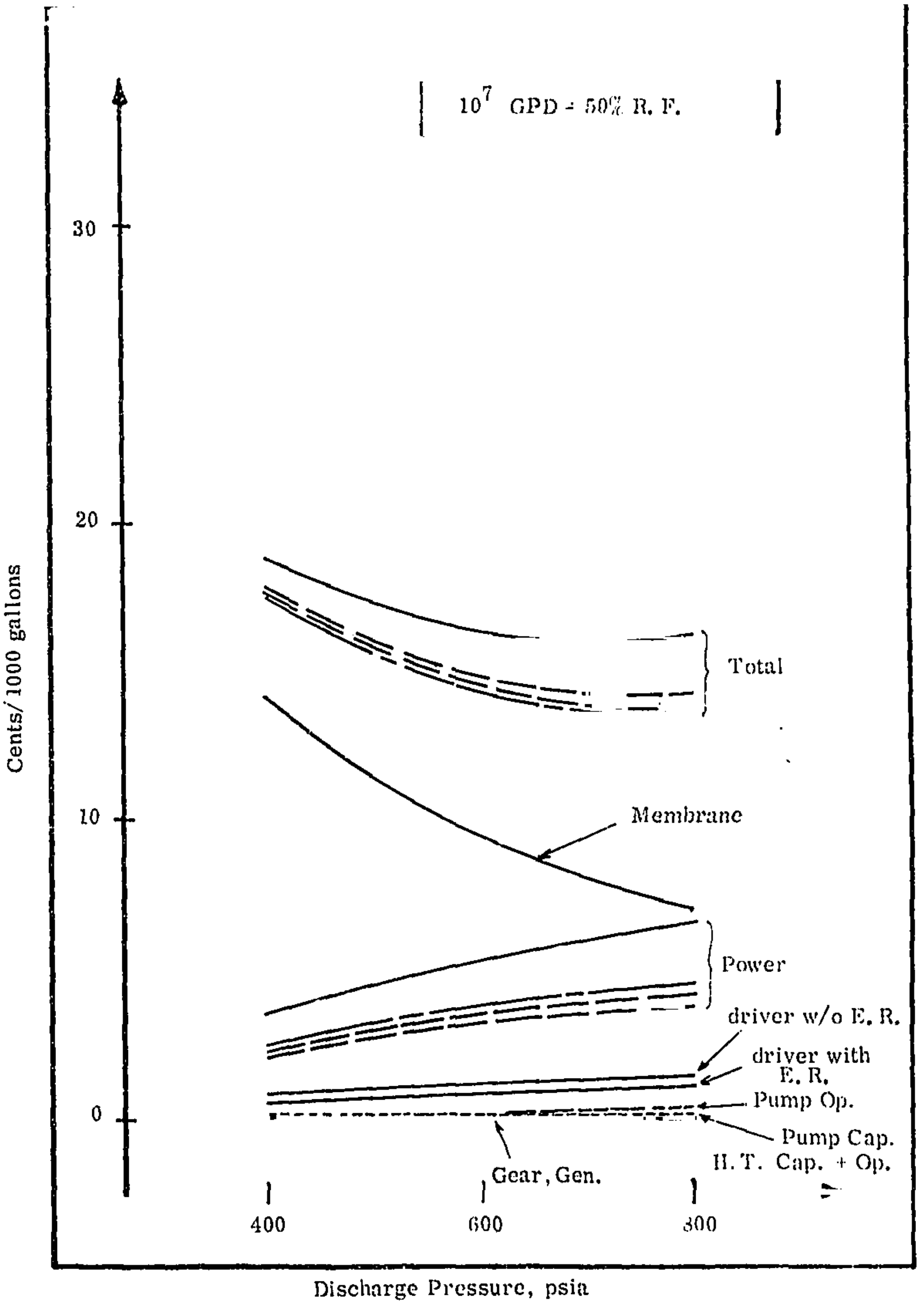


Figure 9.1.37 Contribution to Total Water Cost - Individual Contributions of Capital, Operating and Power Costs Diesel Engine Drive

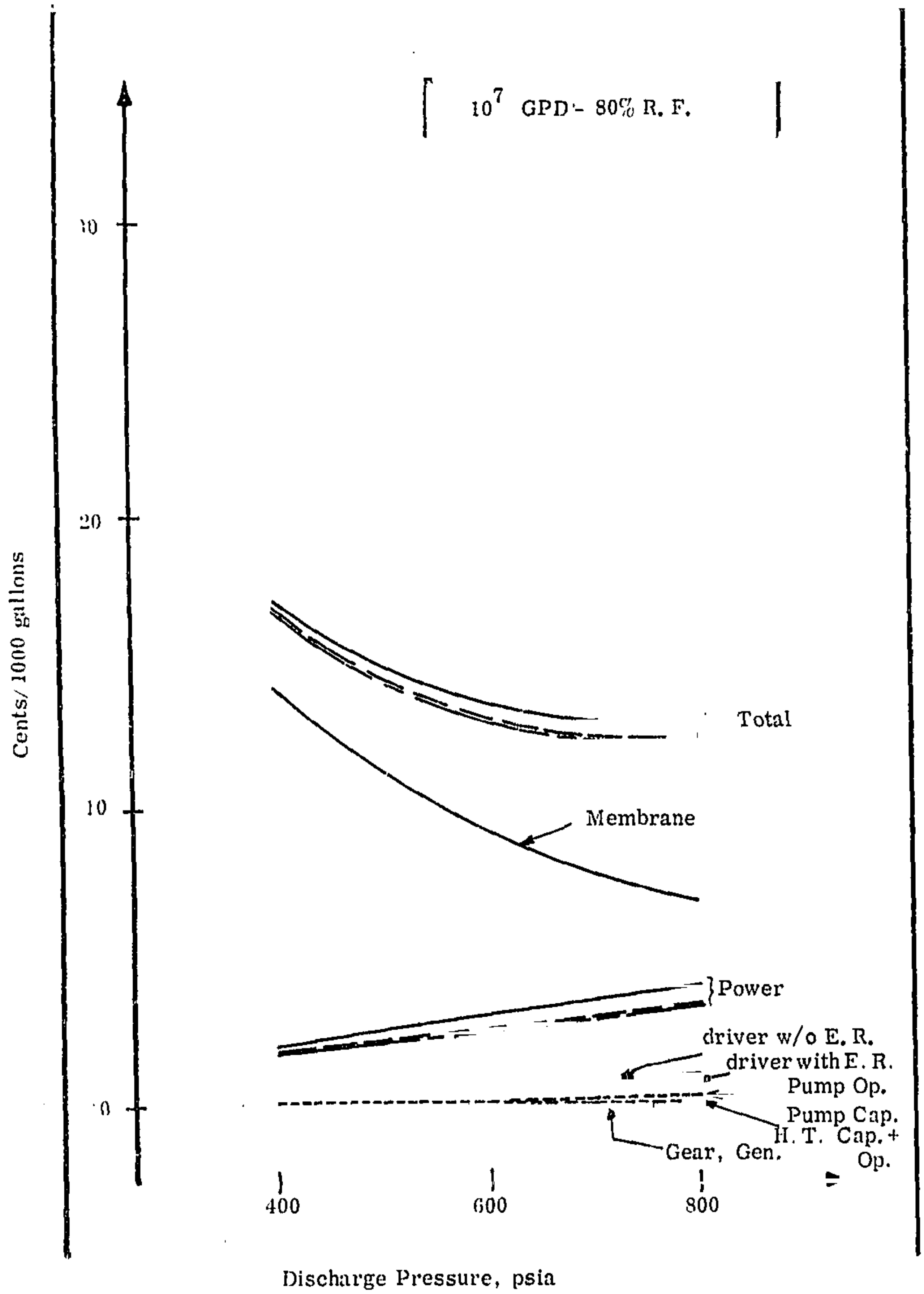
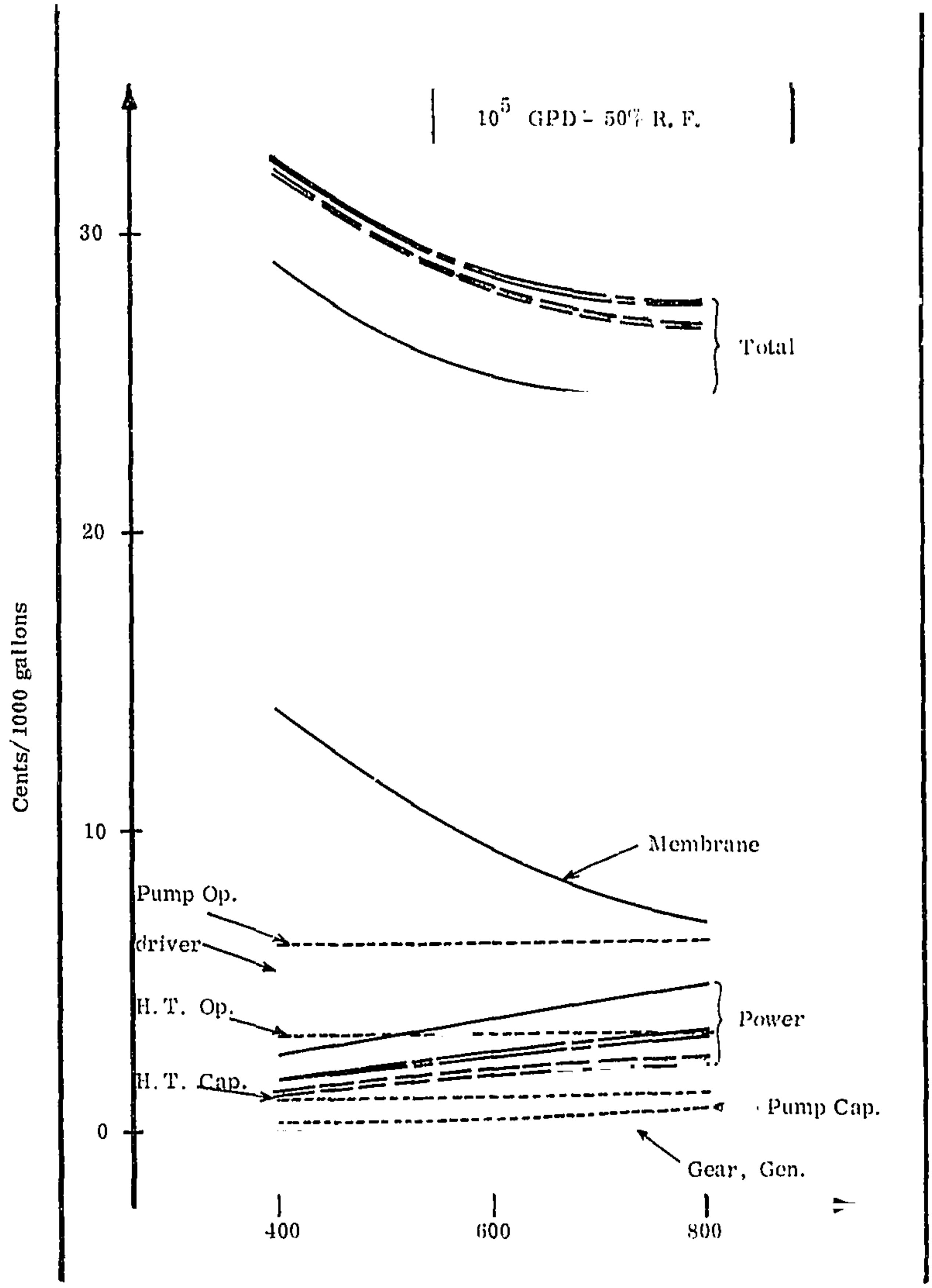


Figure 9. 1. 38 Contribution to Total Water Cost - Individual Contributions of Capital, Operating and Power Costs Diesel Engine Drive



Discharge Pressure, psia

Figure 9.1.39 Contribution to Total Water Cost - Individual Contributions of Capital, Operating and Power Costs Condensing Steam Turbine Drive

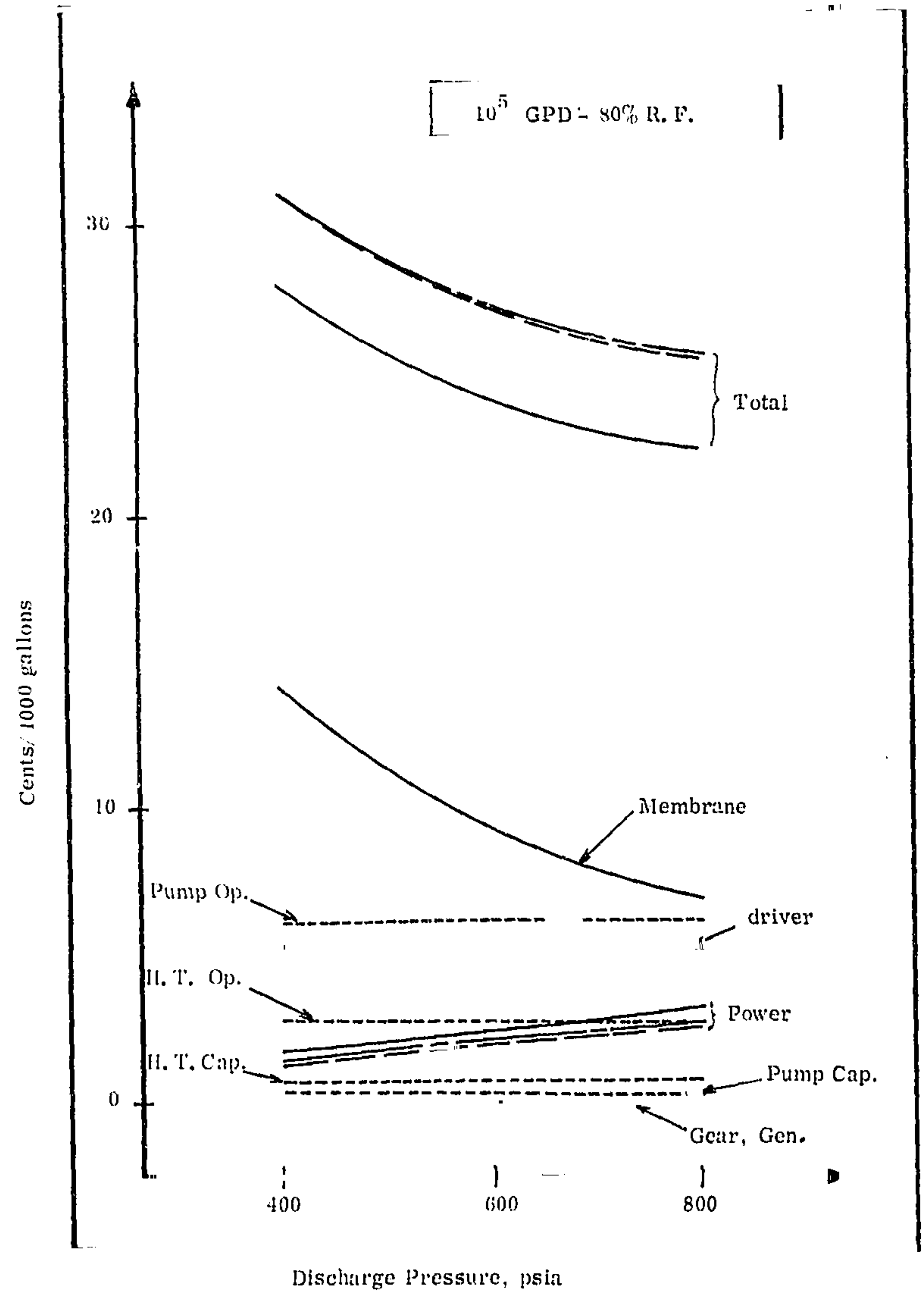


Figure 9. 1. 40 Contribution to Total Water Cost - Individual Contributions of Capital, Operating and Power Costs Condensing Steam Turbine Drive

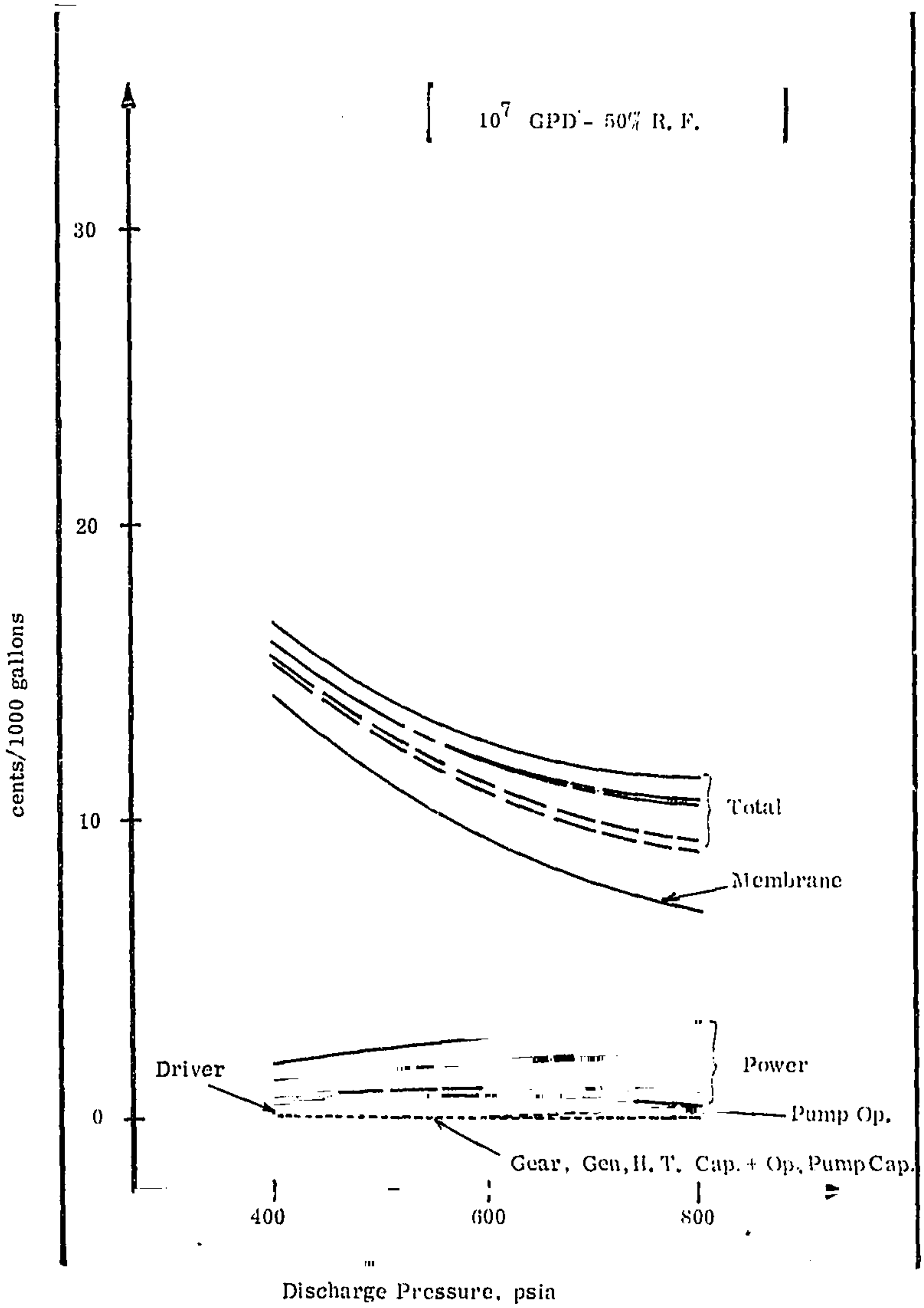
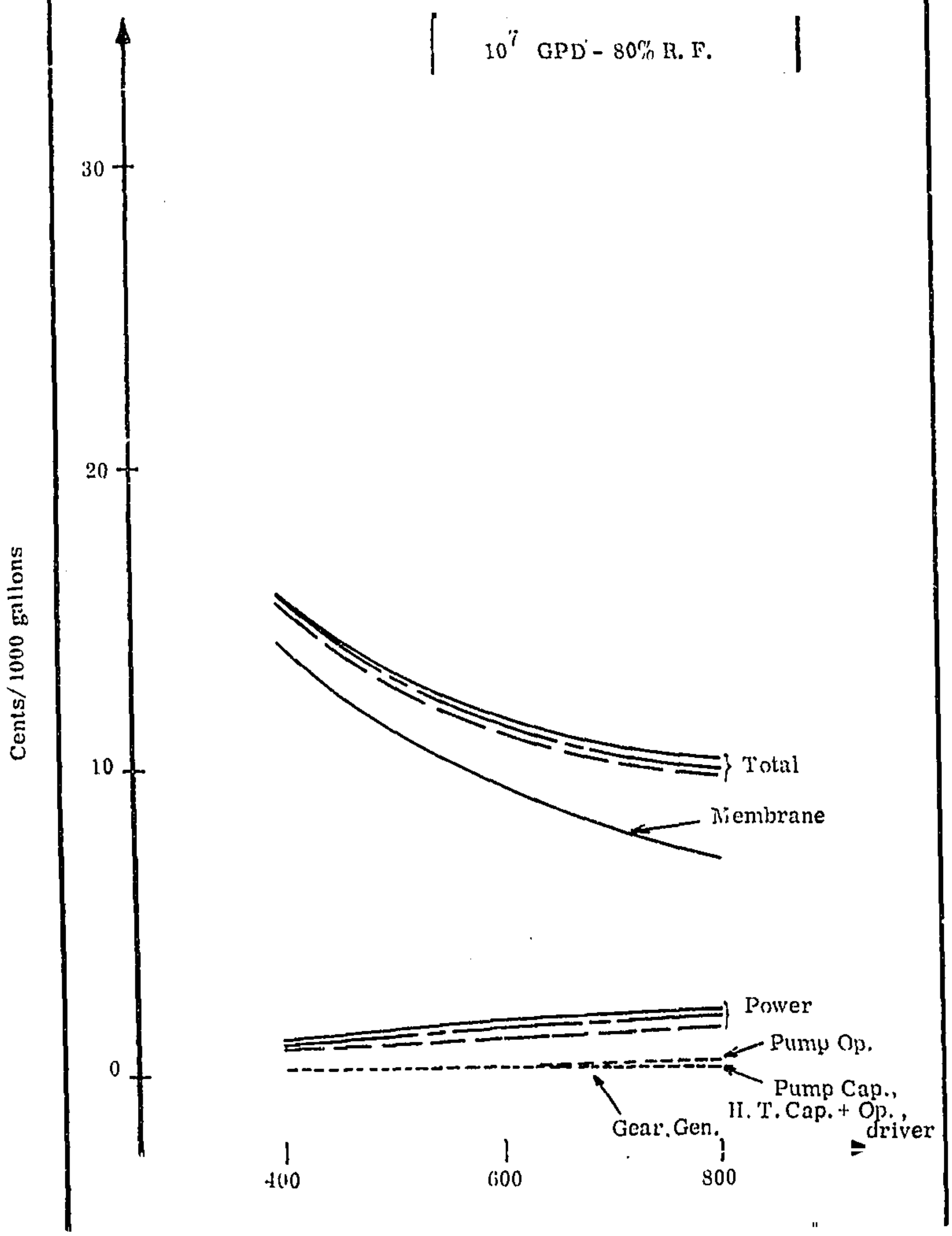


Figure 9.1.41 Contribution to Total Water Cost - Individual Contributions of Capital, Operating and Power Costs Condensing Steam Turbine Drive



Discharge Pressure, psia

Figure 9.1.42 Contribution to Total Water Cost - Individual Contributions of Capital, Operating and Power Costs Condensing Steam Turbine Drive

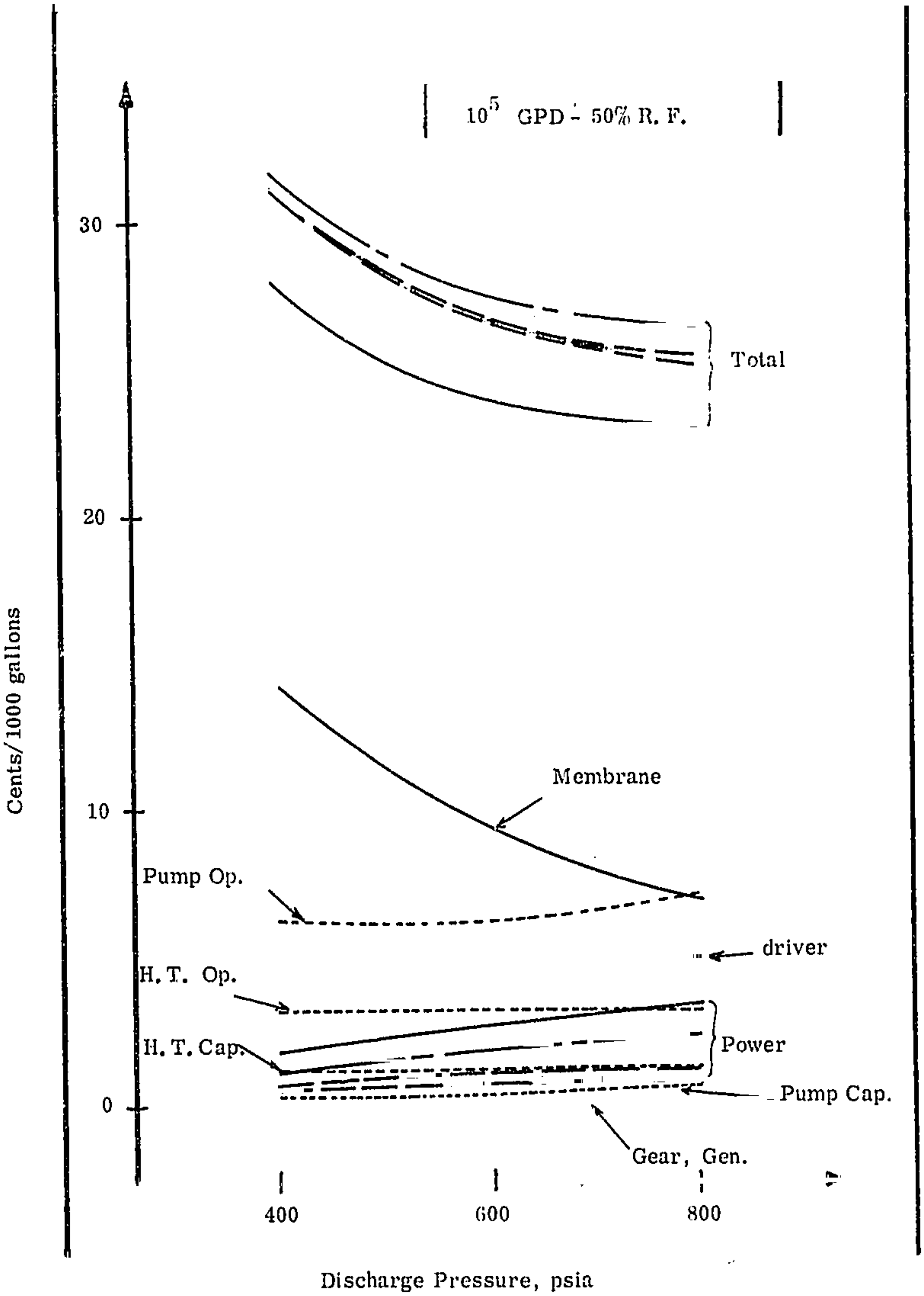


Figure 9.1.43 Contribution to Total Water Cost - Individual Contributions of Capital, Operating and Power Costs Non-Condensing Steam Turbine Drive

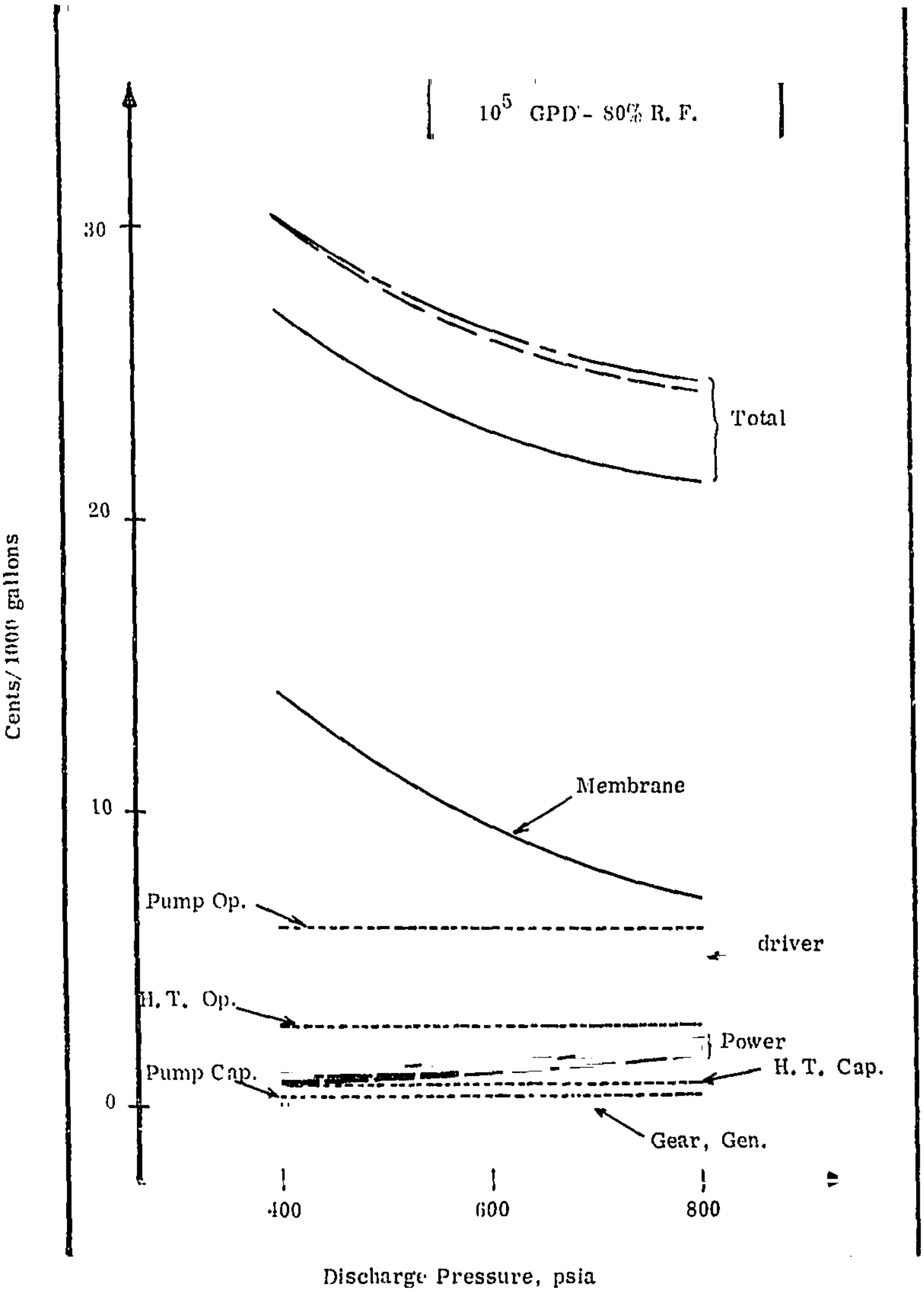


Figure 9. 1. 44 Contribution to Total Water Cost - Individual Contributions of Capital, Operating and Power Costs Non-Condensing Steam Turbine Drive

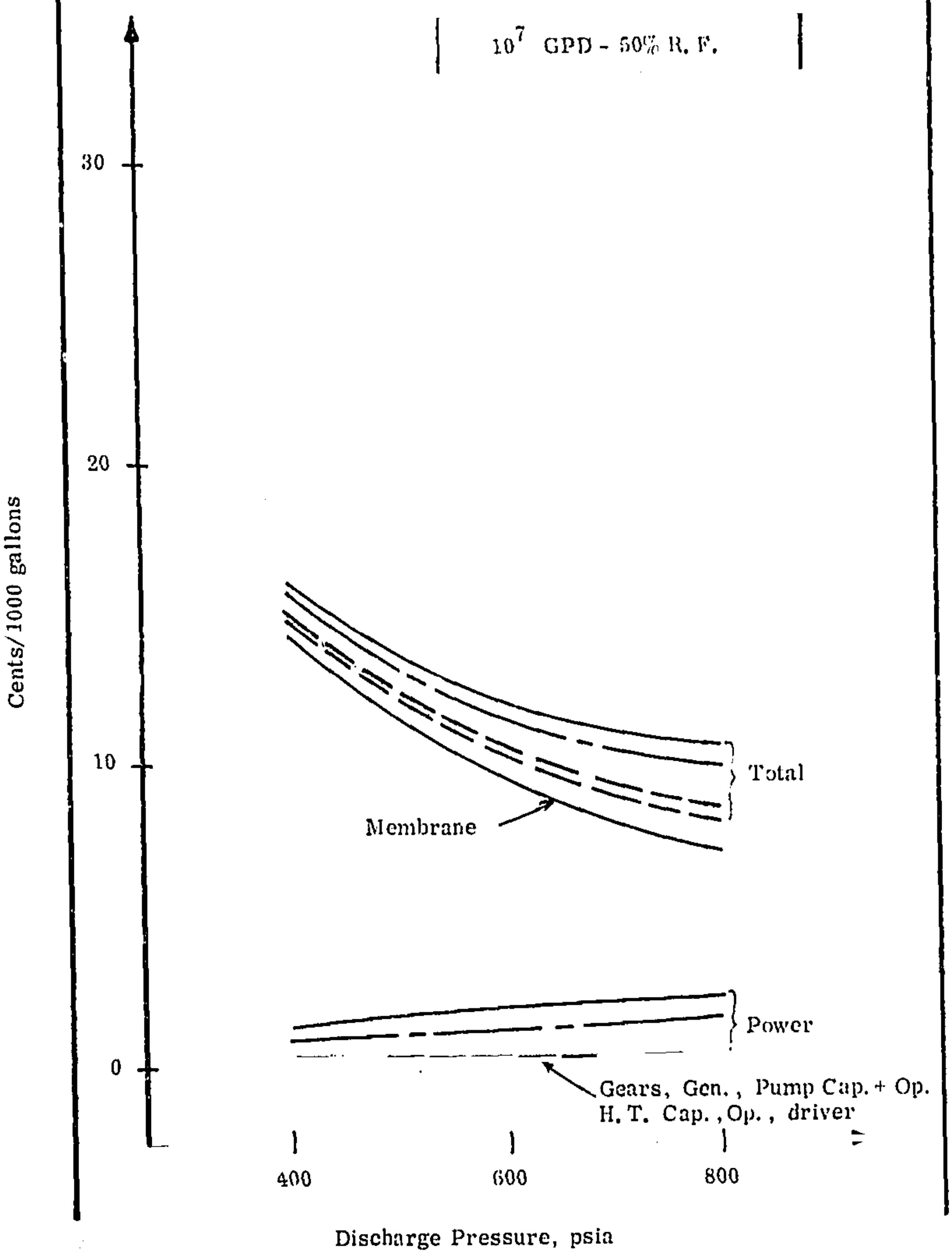
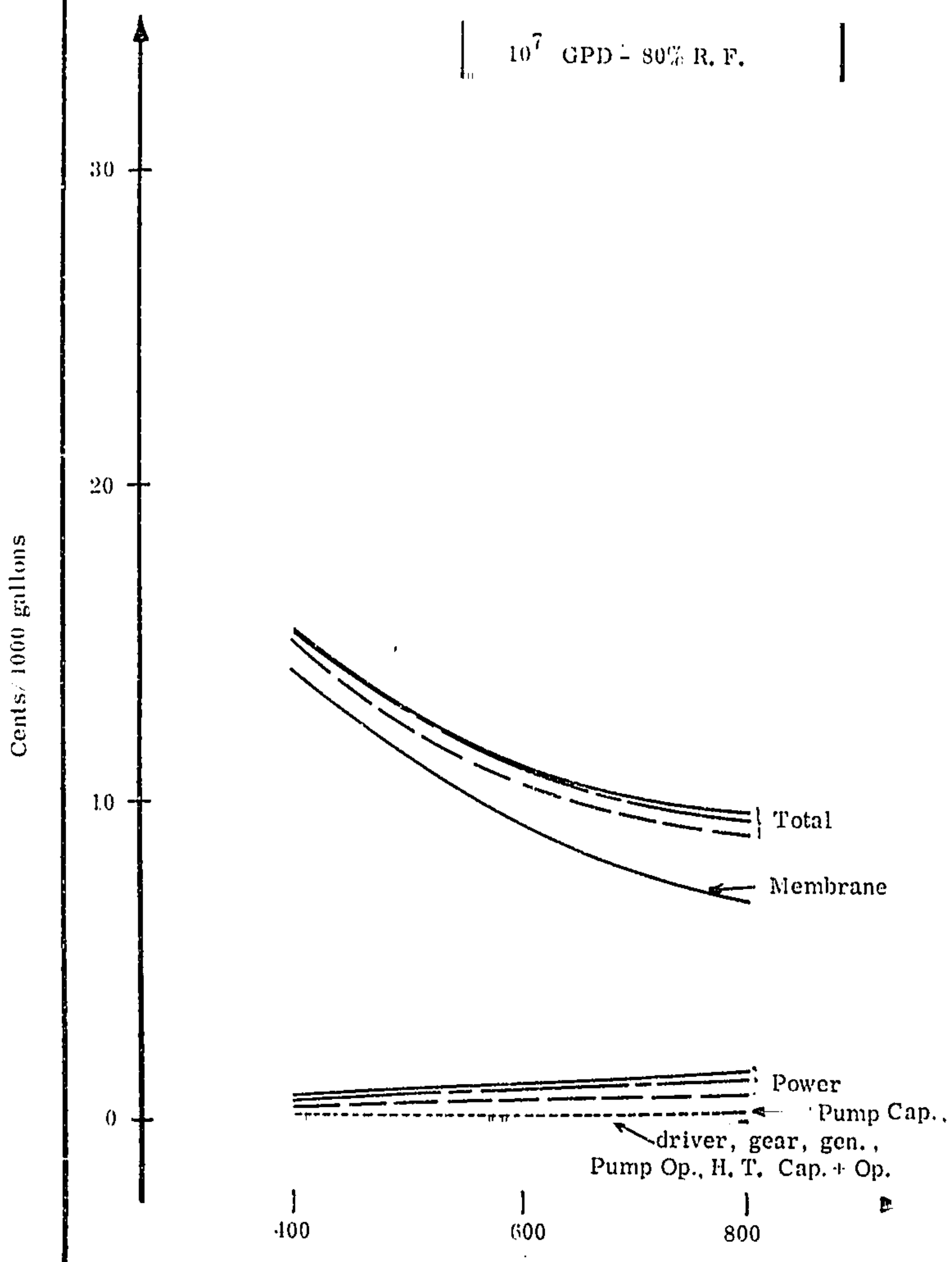


Figure 9.1.45 Contribution to Total Water Cost - Individual Contributions of Capital, Operating and Power Costs Non-Condensing Steam Turbine Drive



Discharge Pressure, psia

Figure 9. 1. 46 Contribution to Total Water Cost - Individual Contributions of Capital, Operating and Power Costs Non-Condensing Steam Turbine Drive

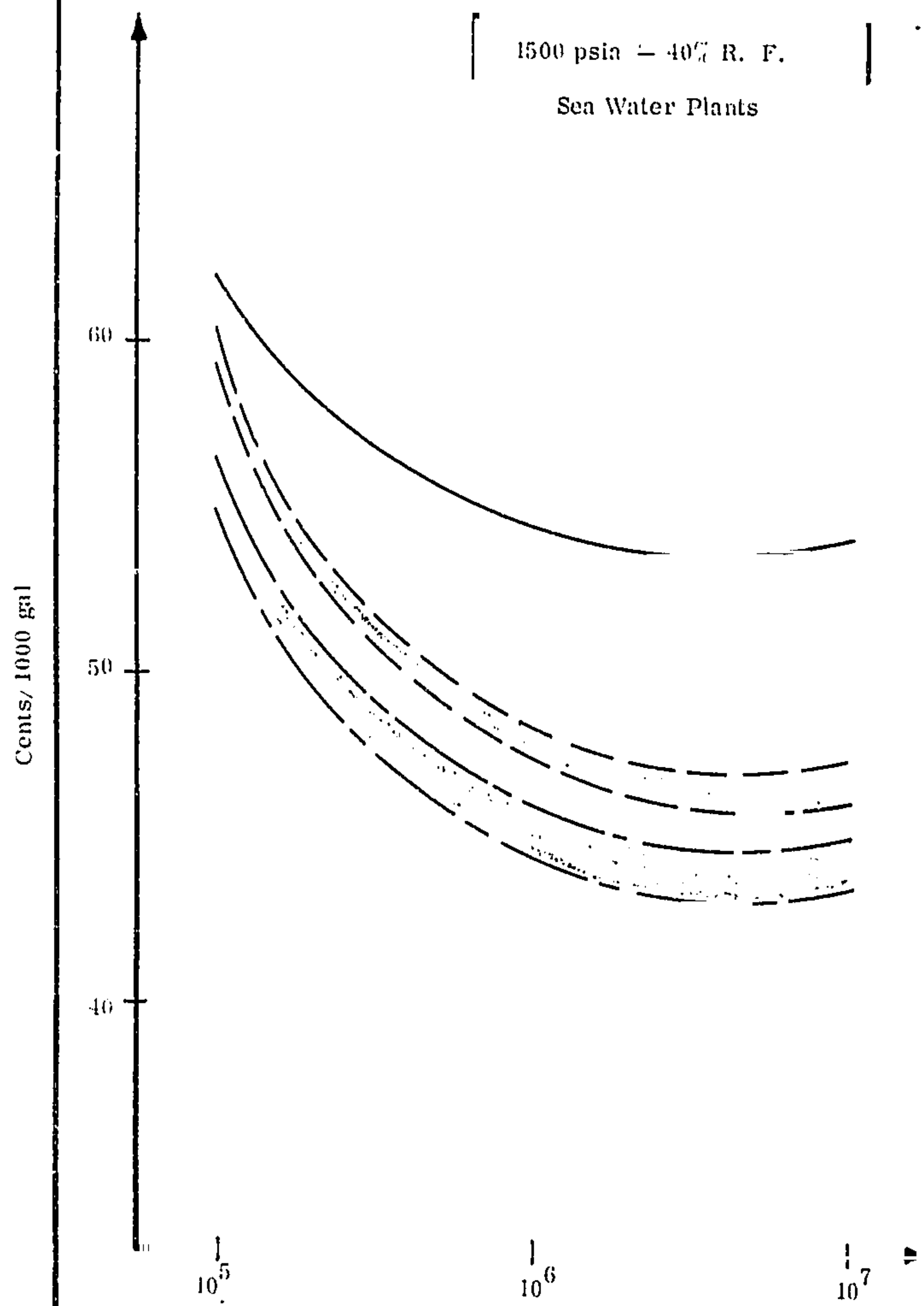
		Figure No.
	Without Energy Recovery	9.1.47 and 9.1.49
	With Electric Generator	
	With Direct H. T. Hook-up	

Type of Driver

		Figure No.
	Electric Motor	9.1.48
	Diesel Engine	
	Steam Turbine	

From Figure 9.1.50 to Figure 9.1.53, all symbols are explained.

Explanation of Symbols Used in Figures
Describing the Contributions to Total Water Cost for Sea Water
Desalination Plants



Fresh Water Output, GPD

Figure 9.1.47 Contribution to Total Water Cost
Electric Motor Driven Plant

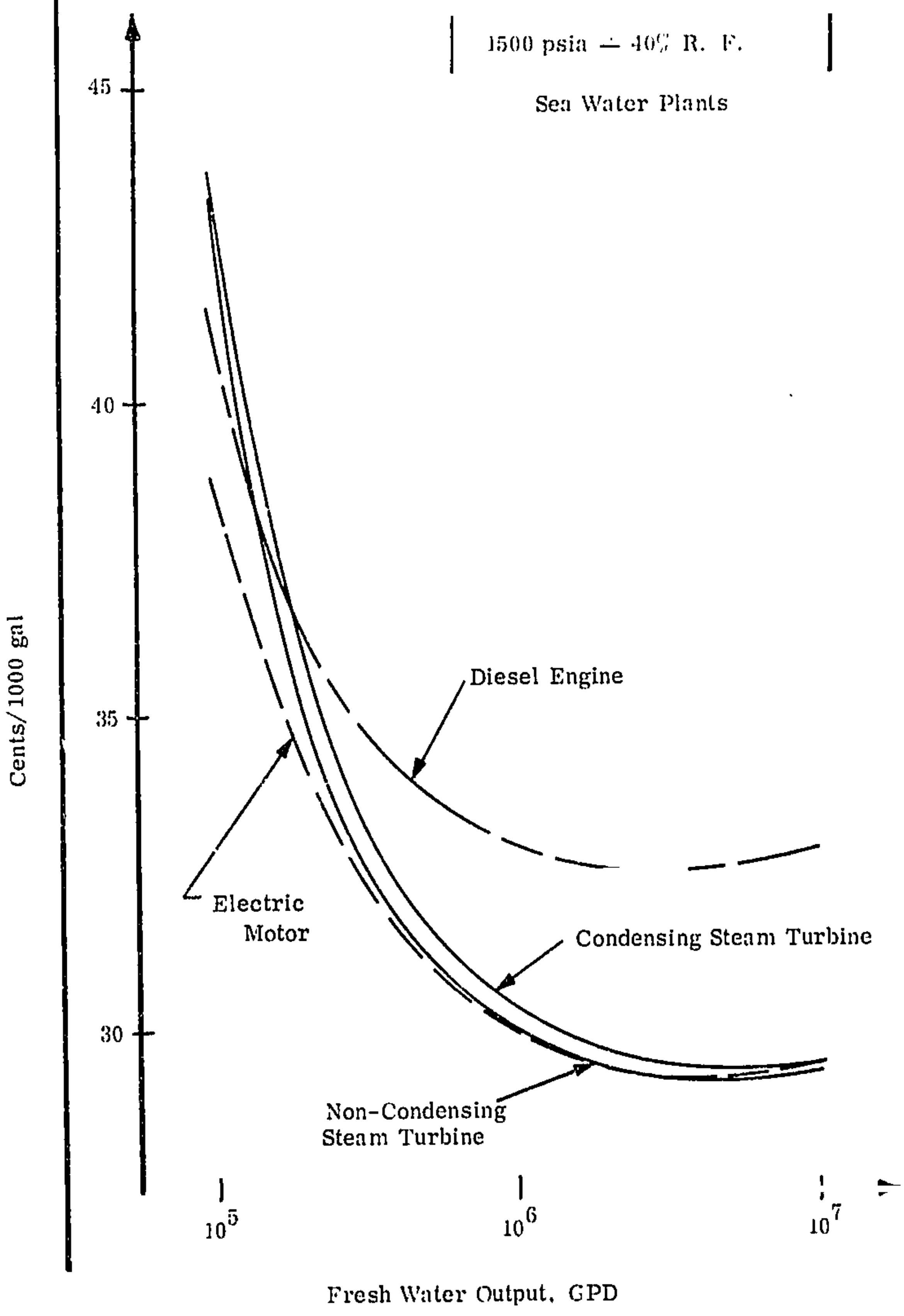


Figure 9.1.48 Contribution to Total Water Cost Exclusive of Power Costs, for Various Drivers

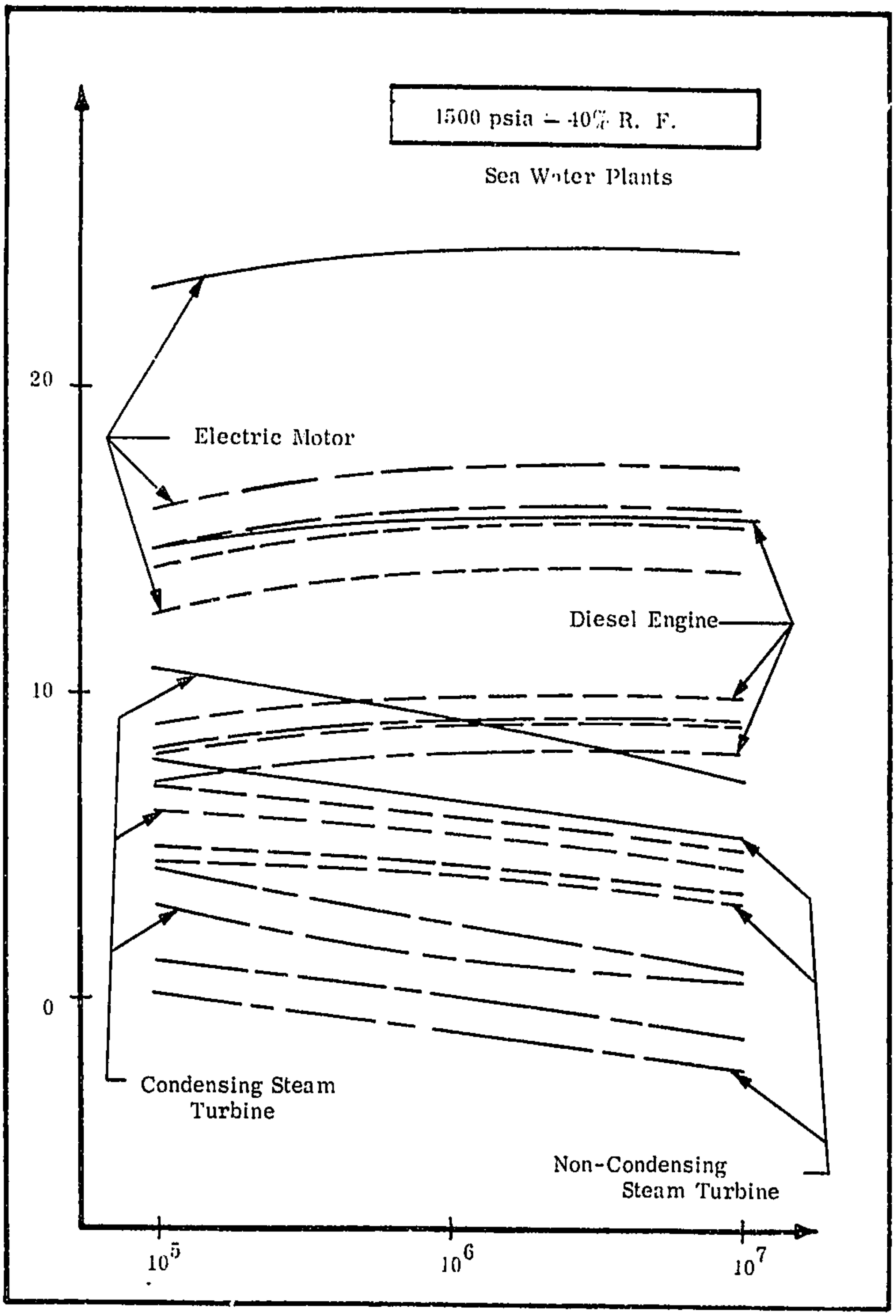


Figure 9.1.49 Power Cost Contribution to Total Water Cost

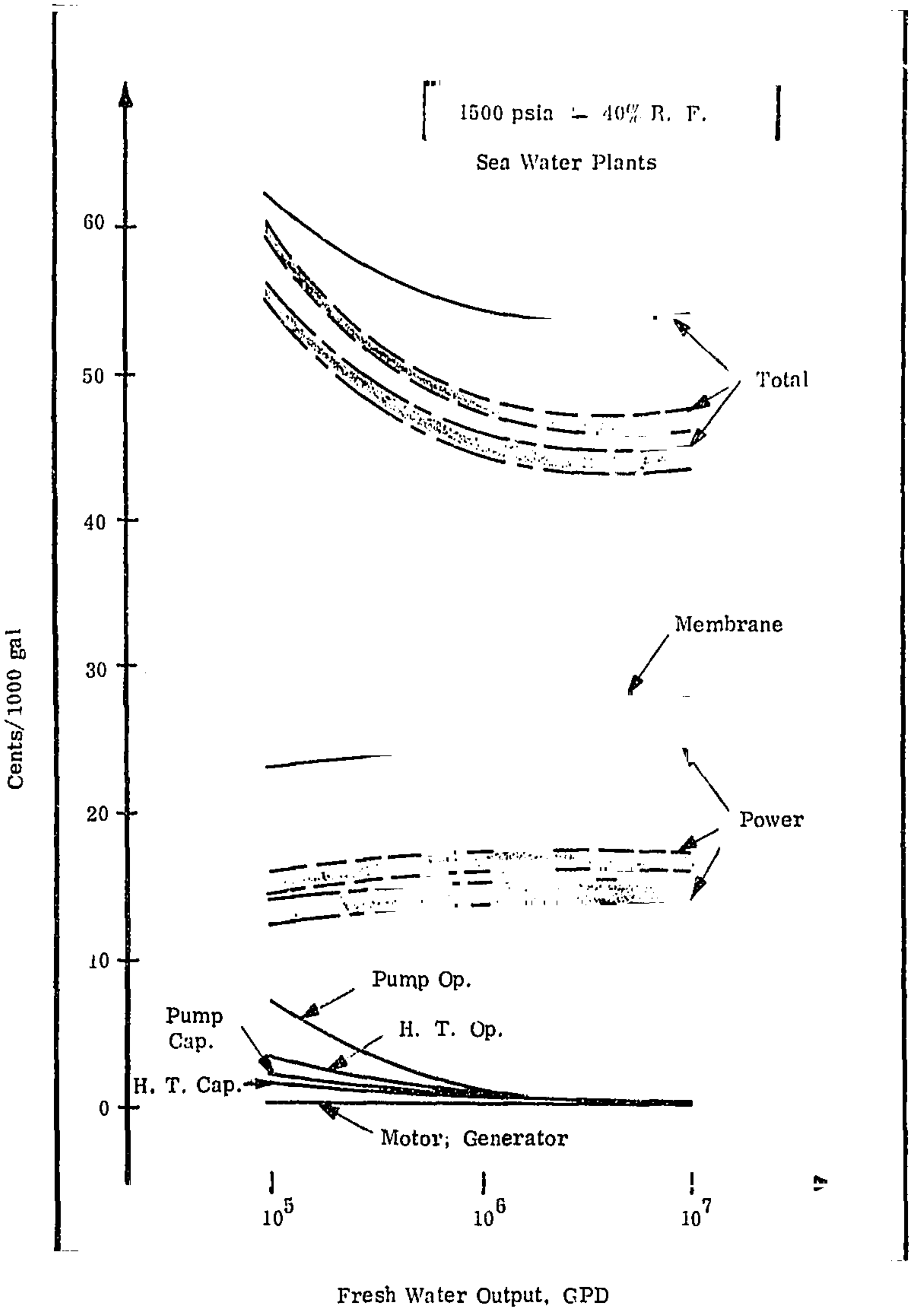


Figure 9.1.50 Contribution to Total Water Cost
Individual Contributions of Capital, Operating and Power Costs
Electric Motor Drive

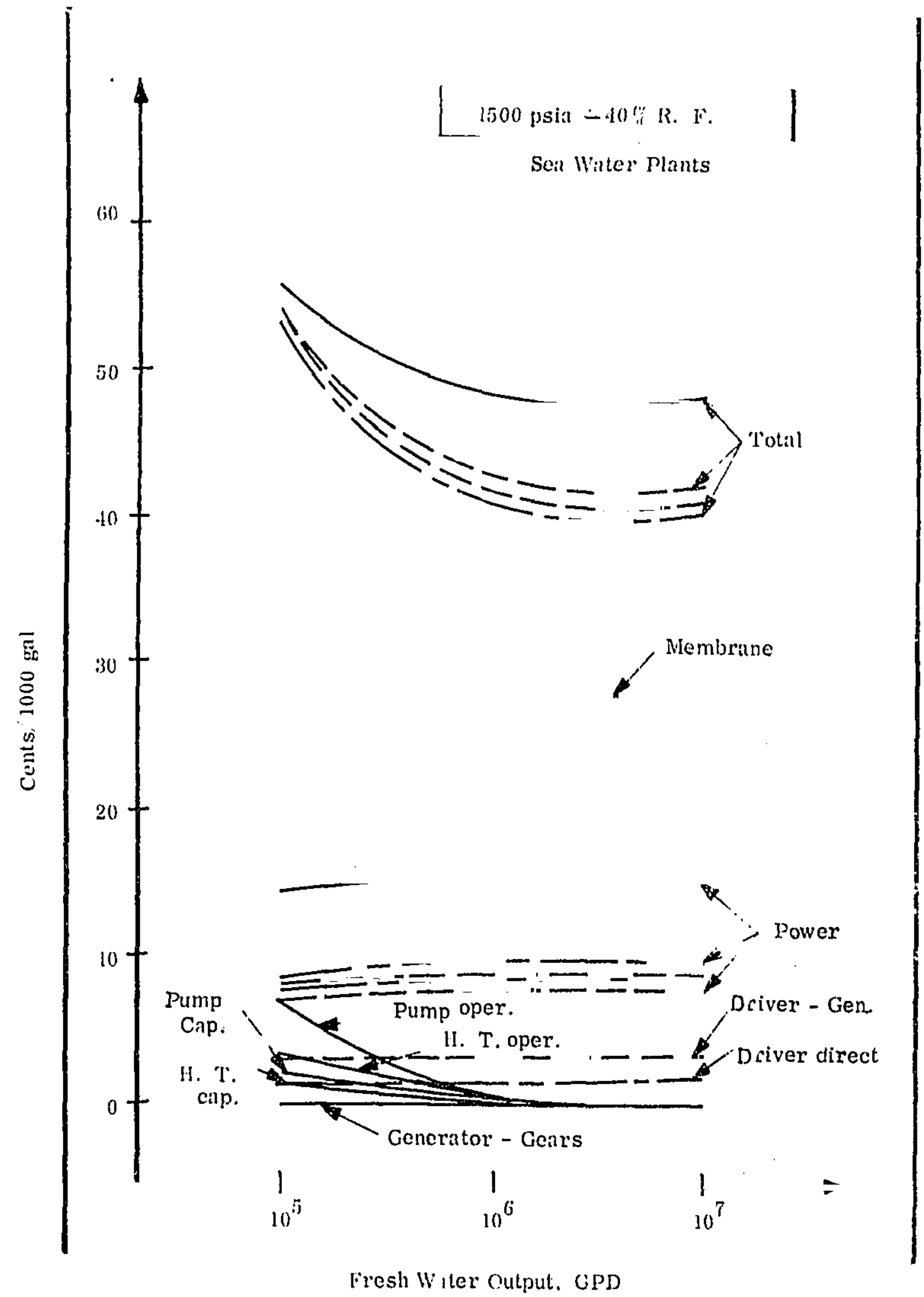
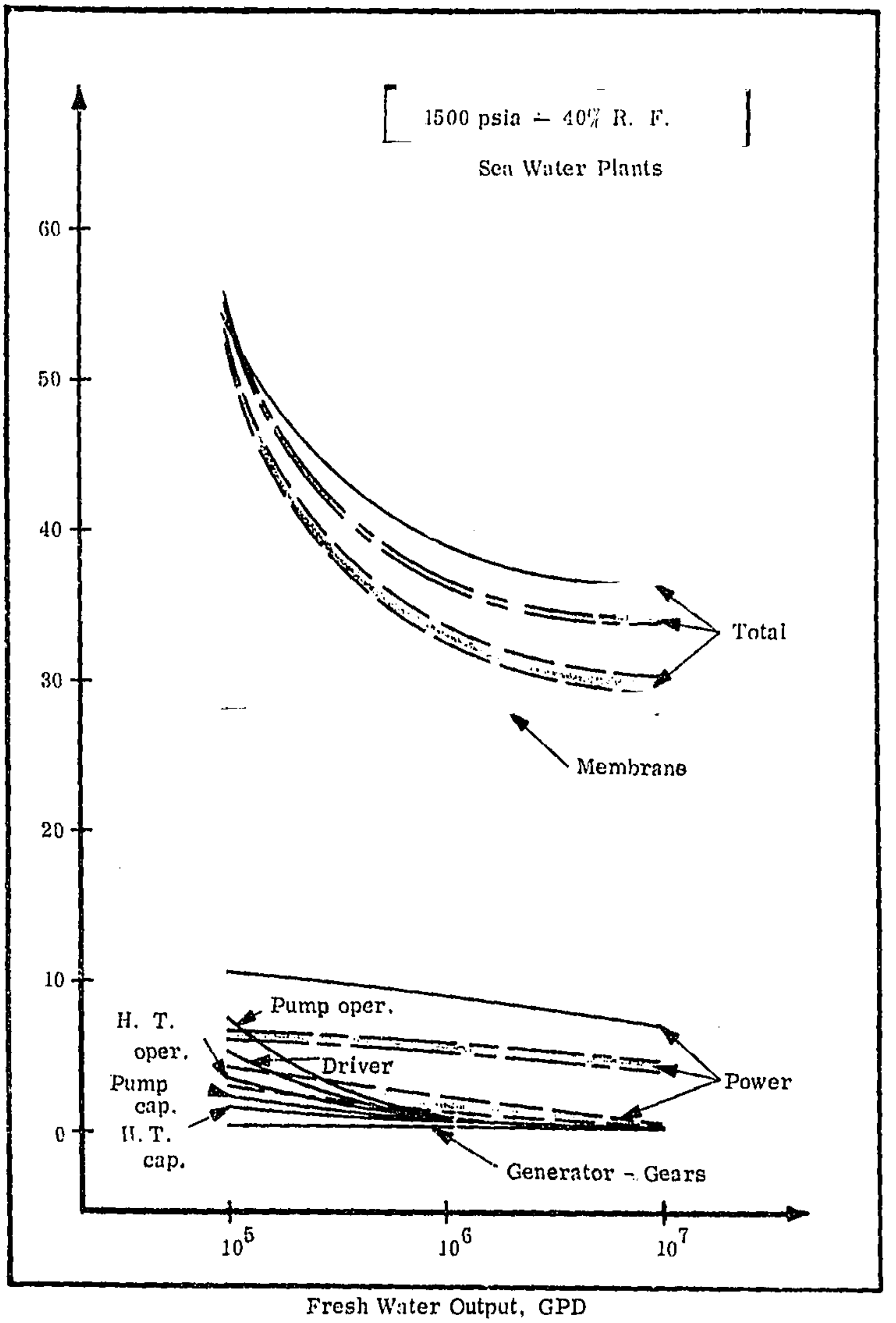


Figure 9.1.51 Contribution to Total Water Cost
Individual Contributions of Capital, Operating and Power Costs
Diesel Engine Drive



Fresh Water Output, GPD

Figure 9.1.52 Contribution to Total Water Cost
Individual Contributions of Capital, Operating and Power Costs
Condensing Steam Turbine

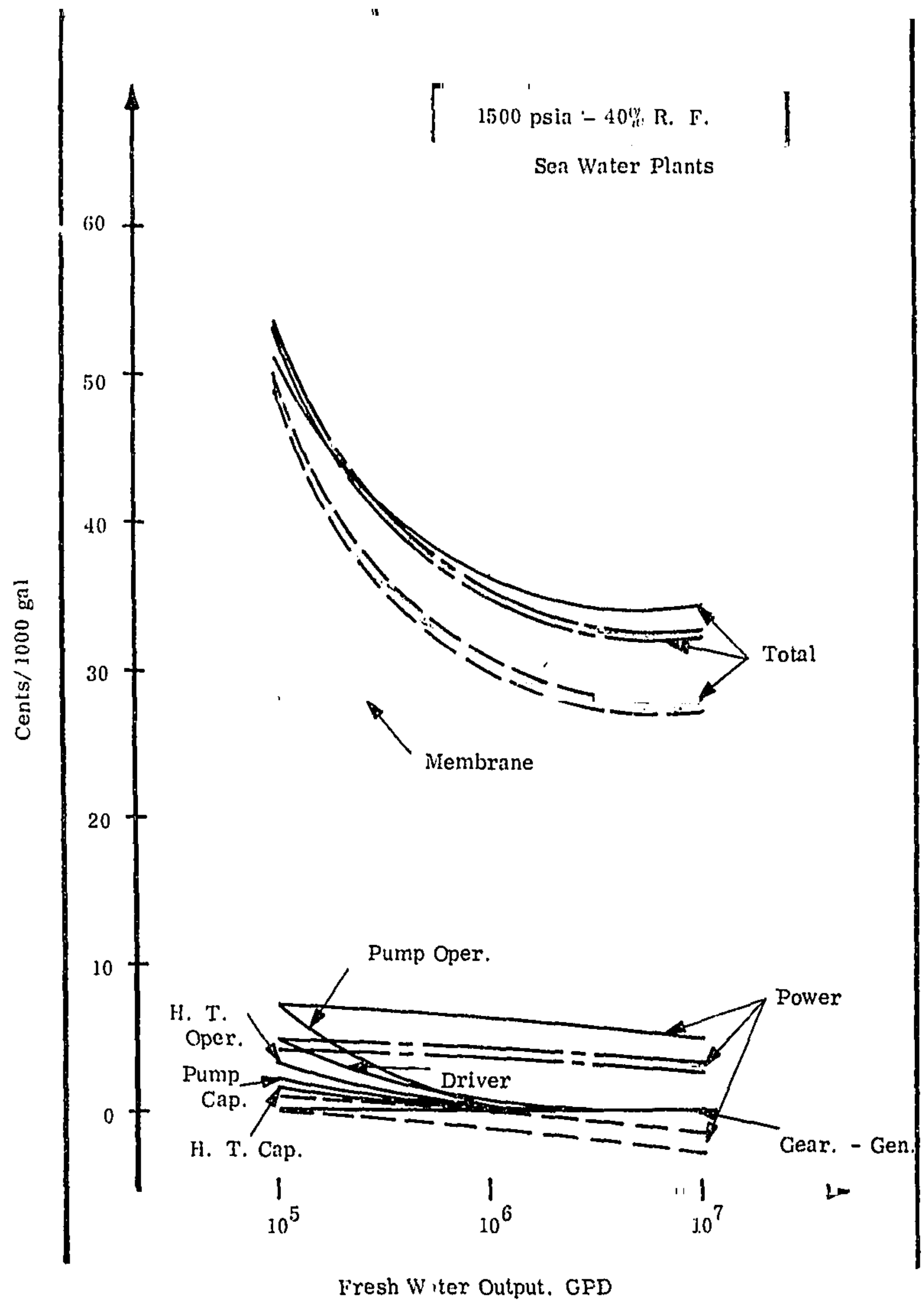


Figure 9.1.53 Contribution to Total Water Cost
Individual Contributions of Capital, Operating and Power Costs
Non-Condensing Steam Turbine

Section 10

CONCLUSIONS AND RECOMMENDATIONS

10.1 Factors Influencing Water Cost

The cost of the power required to drive the high-pressure pump in a reverse osmosis desalination plant is larger than any of the other costs associated with the pumping system. Therefore, increased development costs and increased capital costs can be justified to obtain higher pump efficiencies for reverse osmosis applications.

The cost of various forms of energy - steam, electricity, or diesel fuel - should be carefully evaluated at each plant site to select the most economical type of driver. On the basis of standard energy costs provided by the OSW, steam turbine drivers lead to the lowest pumping system costs for the larger (10^6 GPD to 10^7 GPD) desalination plants. The driver selection is less critical for 10^5 GPD plants.

The use of gears to accept the speed of the most efficient driver and deliver the speed required by the most efficient pump for a given pressure and flow requirement is economically justified. The capital cost added by the gear is generally less than the cost savings achieved by using the most efficient pump and driver combination.

The cost savings which may be obtained by adding a hydraulic turbine to the system depend upon the following parameters in order of descending importance:

1. fresh water production rate
2. fresh water recovery factor
3. membrane pressure level
4. efficiency of the energy recovery system and type of energy recovery arrangement
5. service and maintenance costs (listed as "operating costs")

6. good matching of the various shaft speeds

The brine-side pressure drop in the membrane channels has very little influence on the savings obtained by energy recovery.

10.2 Areas Requiring Further Development

The best materials to use in high-pressure pumps and hydraulic turbines for saline water service have not been determined with certainty. Stainless steel and Ni Resist are recommended by most manufacturers, but their experience with saline water in high-pressure machines is limited. These materials and others should be tested in pumps and turbines to determine operating life characteristics and maintenance requirements. Both centrifugal and reciprocating pumps should be tested; the conclusions obtained for one type are not directly applicable to the other type.

The possibility of developing centrifugal pumps especially for the 10^5 GPD 10^6 GPD plant sizes should be evaluated. The centrifugal pumps which are presently available have been designed for other types of service and may not have the highest efficiencies attainable for desalination plant requirements. High-speed units of smaller size may be more desirable. The manufacturers suggested that higher efficiencies can be obtained by development work to improve impeller and diffuser design and to obtain tighter clearances between stationary and running parts.

There is a need for new pump designs to meet the requirements of the 10^7 GPD plants. Present pump product lines do not extend up to this high-pressure, high-capacity range of operation (Figure 3.2.1).

The reliability of the reciprocating pumps suitable for the 10^5 GPD plant size depends upon the material selection, the valve design, and the seal design. Reciprocating pump manufacturers state that these units are reliable for brackish water. However, some desalination pilot plant tests have uncovered operating problems. Development work on seals, packings, and valves may be desirable.

10.3 Areas Requiring Further Analysis

A number of aspects of the design of pumping systems for the reverse osmosis process require further analysis. These aspects are as follows:

- Equipment Arrangement: Attention should be given to the positioning of the pump, the hydraulic turbine, and the pump driver with respect to the membrane systems and the feed-water pretreatment system. Careful design is desirable to minimize flow losses in connecting piping, to facilitate maintenance, and to reduce the cost of the structure required to support the components.

- Use of Pumps in Parallel: Several lower-capacity pumps operating in parallel can be used to meet the flow requirements of the large size (10^7 GPD or greater) desalination plants. This approach is attractive from the standpoints of plant reliability and efficiency of operation during periods of reduced demand. The technical aspects of parallel operation (controls, maintenance, performance, and reliability) should be analyzed together with the economical aspects.

- Plant Control Techniques: The control system to be used for the pump driver and the power recovery turbine must be designed. Methods to control the pump flow rate and membrane pressure level must be devised. The range of flow rates required for plant operation during peak and minimum demand periods must be established.

Flow control can be obtained in the pumping system by by-passing flow from the pump discharge back to the pump inlet, by varying the number of nozzles and nozzle area in the hydraulic turbine to alter the system resistance, or by varying the pump speed if the driving arrangement permits. An analysis is required to select the best one or best combination of these methods for controlling the pumping system output.

Appendix A

REFERENCE LIST

I. Centrifugal Pumps

1) "Complete Pump Characteristics and the Effects of Specific Speeds on Hydraulic Transients"
B. Donsky - ASME Paper No. 61-Hyd-3.

2) "Centrifugal Pumps"
F. A. Holland & F. S. Chapman - Chemical Engineering, July 1966.

3) "How to Use Specific Speed (as an aid in centrifugal, mixed flow, and axial pump selection)"
F. H. Linley-Machine Design, October 1960.

4) "How to Select Centrifugal Pumps"
H. M. Pollak - Chemical Engineering, February 1963.

5) "The Hydraulic Design of Centrifugal Pumps and Water Turbines"
H. H. Anderson - ASME Paper No. 61-WA-320.

6) "Types of Variable-Speed Pump Controls in Water Service"
A. Ambrosini - AWWA Journal Vol. 56, No. 6, June 1964.

7) "Basic Guides for Better Pump Purchasing"
Purchasing Vol. 46, No. 9, April 27, 1959.

8) "Variable-Speed Pumps--Their Motors and Control"
R. L. Brown - Allis-Chalmers Elec. Rev., Vol. 22, No. 4, 1957.

9) "Mechanical Movement of Fluids"
H. A. S. Gothard - Brit. Power Eng. Vol. 2, No. 4, March 1961.

10) "Problems Posed by Operation of Centrifugal Pumps in Parallel"
Y. de Lachaux - Chaleur & Industrie Vol. 38, No. 387, October 1957.

11) "How Big a Motor ?"
T. W. Edwards - Industry Power Vol. 72, No. 4, April 1957.

12) "Buyer's Guide to European Hydraulic Pumps"
Fluid Power Int. Vol.28, No. 327, June 1967.

13) "Hydraulic Performance Comparison of Centrifugal Pumps"
M. J. Suess - ASCE - Proc. (J. Hydraulics Div.) Vol. 91 No. HY2, March 1965.

14) "Simplified Technique for Estimating Centrifugal Process Pump Costs"
T. J. Sniffen, E. J. Serven - Chemical Engineering Vol. 69, No. 24, November 26, 1962.

15) "New Cost Data for Centrifugal Pumps"
F. S. Chapman & F. A. Holland - Chemical Engineering, Vol. 73, No. 15, July 18, 1966.

16) "Centrifugal Pump Selection"
E. Keller - ASHRAE J. Vol. 5, No. 3, March 1963.

17) "Centrifugal Pumps and Rotative Speed"
E. J. Serven - Chemical Engineering Vol. 70, No. 7, April 1, 1963.

18) "Water-Pressure Oscillations in Volute Casings of Storage Pumps"
Y. N. Chen - Sulzer Tech. Rev.--Research No. 1961.

19) "Specialized Centrifugal Pumps"
The Engineer, Vol. 217, No. 5649, May 1, 1964.

20) "How Far Can Centrifugal Pump Flow Be Throttled ?"
J. R. Broce - Power Vol. 107, No. 11, November 1963.

II. Centrifugal Pumps - Installation, Operation, Maintenance

21) "Wearing Rings and Centrifugal Pump Efficiency"
N. Aradvilentchuk - ASME Paper 62-MPE-2, May, 1962.

22) "New Search for Criteria of Equipment Reliability Evaluation"
I. J. Karassik - ASME Paper No. 59-SA-2, June, 1959.

23) "Care and Maintenance of Centrifugal Pumps"
W. Wessing - Engrs' Digest Vol. 18, No. 11, November 1957.

24) "Operation and Maintenance of Centrifugal Pumps"
E. J. Watts - AWWA Journal Vol. 54, No. 6, June 1962.

25) "Installation of Centrifugal Pumps"
I. J. Karassik, R. Carter - Air Conditioning, Heating & Vent. Vol. 55, No. 7, July 8, 1958.

26) "Operation of Centrifugal Pumps"
I. J. Karassik, R. Carter - Air Conditioning, Heating & Vent. Vol. 55, No. 9, September 10, 1958.

27) "Maintenance and Repair. Maintenance of Centrifugal Pumps"
I. J. Karassik, R. Carter - Air Conditioning, Heating & Vent. Vol. 55, No. 11, November 12, 1958.

28) "Check Your Centrifugal Pumps"
E. L. Brooks - Industry Power Vol. 73, No. 4, September 1957.

29) "Centrifugal Pump Troubles"
I. J. Karassik, R. Carter - Air Conditioning, Heating & Vent. Vol. 56, No. 2, February 3, 1959.

30) "We Saved $7000 by Repairing This Old Centrifugal Pump"
J. Murphy - Power Vol. 101, No. 6, June 1957.

31) "Prevent Pump Headaches"
C. R. McFarland - Power Vol. 100, No. 10, October 1956.

32) "Centrifugal Pumps--Correct Installation and Starting Give Them Good Lease on Life"
E. Schwandt - Plant Eng. Vol. 11, No. 8, August 1957.

33) "Selective Maintenance of Hydraulic Systems"
L. S. McNickle, Jr. - Coal Age Vol. 67, No. 11, November 1962.

34) "Mechanical Seals for Nonlubricating Hydrocarbons"
A. L. Decker - ASME Paper No. 57-PET-2, September, 1957.

III. Positive Displacement Pumps

35) "Reciprocating Pumps Bid for New Jobs"
J. W. McConaghy - Power Vol. 107, No. 11, November 1963.

36) "Pumps--Gear, Vane and Piston"
E. R. Carpenter - National Conference on Indus. Hydraulics - Proc. Vol. 16, 1962.

37) "New Look at Piston Pump Design"
R. L. Ruth - Nat. Conference on Indus. Hydraulics -- Proc. Vol. 16, 1962.

38) "Positive Displacement Pumps"
F. S. Chapman & F. A. Holland, Chemical Engineering, Vol. 73, No. 4, February 14, 1966.

IV. Metering Chemical and Special Pumps

39) "Chemical Feed Pumps"
M. M. Kirk - Chem. Eng. Vol. 72, No. 5, March 1, 1965.

40) "Hydraflo Pump"
Water & Water Eng. Vol. 63, No. 758, April 1959.

41) "Air-Lift Pump--Theory and Optimization"
D. J. Nicklin - Instn. Chem. Engrs.--Trans. Vol. 41, No. 1, 1963.

42) "Metering Pumps"
F. D. Marton - Instruments & Control Systems Vol. 36, No. 4, April 1963.

43) "Kingsbury Slipper Pump...Design for Higher Pressures"
J. M. Roth, D. L. Roth - Applied Hydraulics & Pneumatics Vol. 13, No. 4, April 1960.

44) "Airhydropump: High Pressure Intensifier"
B. Crossland, R. L. Alexander - Engineering Vol. 189, No. 4902, April 1, 1960.

45) "Pumps for Chemical Industry"
Chem. Age Vol. 86, No. 2207, October 28, 1961.

46) "Characteristics and Applications of Water-Jet-Pump"
R. Silvester - Houille Blanche Vol. 16, No. 4, Aug.-Sept. 1961.

47) "New Equipment and Processes in Fluid Handling"
Brit. Power Eng. Vol. 2, No. 4, March 1961.

48) "Hydraulic Pumps and Their Applications"
P. J. C. Lowe - Fluid Power Int., Vol. 28, No. 327, June 1963.

V. Boiler Feed Pumps. Boiler Feed Pump Turbines

49) "Recent Developments in Boiler Feed Pumps"
R. A. Strub, W. Rymann, Sulzer Tech. Rev.--Research No. 1961.

50) "How to Select Boiler-Feed Pumps for Modern Industrial Power Plant"
I. J. Karassik - Power Vol. 106, No. 2, February 4, 1962.

51) "New Boiler Feed Pump Designs in U.S. and Europe Show Improved Reliability"
J. J. Jacobs - Power Vol. 109, No. 4, April 1965.

52) "How Do You Operate Your BF Pumps"
I. J. Karassik - Power Eng. Vol.61, No. 3, March 1957.

53) "Recent Developments of Central Station Boiler Feedwater Pumps"
W. L. Wathen - Power Eng. Vol. 67, No. 10, October 1963.

54) "Boiler Feed Pump Turbines--Some Aspects of Their Design and Economics"
L. A. Holt - South African Mech. Engr. Vol. 13, No. 9, April 1964.

55) "Designers Tackle Automation of Boiler Feed Pumps"
I. J. Karassik - Elec. Light & Power Vol. 41, No. 7, July 1963.

56) "Boiler Feed Pump Automation"
I. J. Karassik - Power Eng. Vol. 65, No. 3, March 1961.

57) "Steam Power Plant Clinic"
I. J. Karassik - Combustion Vol. 30, No. 2,3,4,5, Aug. 1958.

58) "Seven Years' Experience with Single Turbine-Driven Feed Pumps Serving Large Generating Units"
T. T. Frankenberg, J. A. Tillinghast - Combustion Vol. 36, No. 1, July 1964.

59) "Auxiliary Turbine Drives for Boiler Feed Pumps"
A. W. Rankin - Combustion Vol. 35, No. 7, January 1964.

60) "Combination BFP Drive Economical for Big Unit"
A. W. Rankin - Elec. World Vol. 160, No. 19, November 4, 1963.

61) "Turbines Gain as BF-Pump Drives for Large Units but Motors Still Hold Lead"
L. M. Olmsted - Elec. World Vol. 149, No. 11, March 17, 1958.

62) "Where Do Boiler Feed-Pump Turbines Stand Today ?"
H. A. Mayor - Elec. World Vol. 154, No. 8, August 22, 1960.

63) "Turbines Stage Come-Back for Large B-F Pump Drives"
J. F. Ransom, E. L. Pace - Power Eng. Vol. 60, No. 10, October 1956.

64. "Economics of Main-Turbine Drives for Central-Station Boiler-Feed Pumps"
I. J. Karassik - Power Vol. 100, No. 11, November 1956.

65) "World's Largest Feed Pump is Driven by Main Turbine"
T. Hall - Power Vol. 100, No. 10, October 1956.

66) "A Comparative Engineering Analysis of Feed Pump Drives"
F. A. Ritchings, Jr., M. W. Larinoff - Consulting Engr. Vol. 11, No. 6, December 1958.

67). "Boiler Feed Pump Drives"
J. H. Goodell, P. Leung, ASME - Paper 64 - PWR - 7 for meeting September 27- October 1, 1964.

68) "Can Hydraulic Couplings Double as Feedwater Regulators ?"
I. J. Karassik - Combustion Vol. 29, No. 6, December 1957.

69) "Prevent Flashing in Boiler Feed Pump Leakoff Lines"
A. R. Bush - Power Eng. Vol. 66, No. 4, April 1962.

70) "Large Boiler Feed Pump Practice and Design"
G. F. Arkless - Brit. Power Eng. Vol. 4, No. 4, March 1962.

71) "Recent Boiler Feed Pump Trends"
I. J. Karassik - Steam Engr. Vol. 28, No. 326, December 1958.

72) "Development of High Pressure Centrifugal Pumps for Controlled Circulation and Supercritical Steam Generators"
R. E. Allen - ASME Paper No. 57-A-243, December, 1957.

73) "Design of Modern Boiler Feed Pumps"
H. H. Anderson - Instn. Mech. Engrs.-Proc. Vol. 175, No. 12, 1961.

74) "Some Problems Associated with Development of Modern Boiler Feed Pumps"
G. W. Heslett - Instn. Engrs., Australia-Journal Vol. 33, No. 10, Oct.-Nov. 1961.

75) "Changing Philosophy in Design, Application and Operation of High Pressure Boiler Feed Pumps"
I. J. Karassik - Instn. Engrs. Australia-Journal Vol. 34, No. 12, December 1962.

76) "How Big a Boiler Feed Pump Motor?"
I. J. Karassik - Power Eng. Vol. 63, No. 2, February 1959.

77) "Variable-Speed Control as Applied to Water Pumping"
M. H. Owen - Am. Water Works Assn.-Journal Vol. 50, No. 5, May 1958.

78) "Electric Drives for Pumping Plant"
J. Lownie - Instn. Certificated Engrs. S. Africa-Journal Vol. 32, No. 6, June 1959.

79) "Matching Boiler Feed Pump to Your Job"
R. A. Miller - Allis Chalmers Elec. Rev. Vol. 23, No. 4, 1958.

80) "Drive of Feed Pumps by Condensing Steam Turbines"
R. Hohl - Escher Wyss News Vol. 34, No. 1, 1961.

81) "Steam Turbine Returns to Drive BF Pumps at Huntley, Replacing Motors."
R. P. Moore - Elec. World Vol. 149, No. 11, March 1958.

82) "Selecting the Largest Feed Pump Drive."
Power - Vol. 100, No. 10, October 1956.

83) "Turbine-Driven Boiler Feed Pumps Respond to Precise Control."
M. Gottlieb, M. Pierpoline - Elec. World Vol. 161, No. 14, April 1964.

84) "Integration of Turbine-Driven BFP in Large Power Plants."
E. L. Pace - ASME Paper No. 59-A-196.

VI. Foreign Papers on Boiler Feed Pumps and Boiler Feed Pump Drives

85) "Efficiency Study of Types of Feedwater Pump Drive in Large Power Plants."
V. Ya. Ryzhkin, E. P. Volkov - Teploenergetika No. 10, Oct. 1963.

86) "Selection on Drive for Boiler Feed Pumps"
R. Mäule - Brennstoff-Waerme-Kraft Vol. 13, No. 12, Dec. 1961.

87) "Determination of Most Economical Type of Variable-Speed Drive for Boiler Feedwater Pumps"
P. Haack-Siemens Zeit. Vol. 38, No. 7, July 1961.

88) "Experiments for Correcting Efficiency of Boiler Feed Pumps"
W. Gruenendiek - Brennstoff-Waerme-Kraft Vol. 15, No. 12, December 1963.

VII. Pump Seals

89) "Select Best Pump Seal"
H. E. Tracy - Chemical Engineering - Vol. 64, No. 4, April 1957.

90) "Floating Ring Seals - Design and Application. Special Reference to High Pressure."
Leonard H. Sence - ASME Paper No. 63 AHGT-31

91) "Development of Floating Ring Type Stuffing Boxes for Eddystone Boiler Feed Pumps."
A. Brkich, R. E. Allen - ASME Paper No. 59-A-259

VIII. Hydraulic Turbines. Pumps Used as Hydraulic Turbines

92) "Construction of Francis Turbines in Norway."
H. Christie - Teknisk Ukeblad Vol. 111, No. 10, March 5, 1964.

93) "The 66,300 kw, 465 m High Head Francis Turbine"
T. Arikawa, S. Ito, Toshiba Rev. No. 21, Spring 1965.

94) "Installation and Operating Experiences with Kemano 2500 Foot Head Impulse Turbines."
J. T. Madill, F. P. Gordon - Eng. Journal Vol. 41, No. 2, Feb. 1958.

95) "Characteristics of Francis Turbines"
D. Zanobetti - Water Power Vol. 11, No. 3, March 1959.

96) "Novel Hydraulic Motors in 1960"
M. Medici - Elettrotecnica Vol. 48, No. 4, April 1961.

97) "Hydraulic Turbine Development during the Last Few Years"
A. Puyo - Houille Blanche Vol. 18, No. 1, Jan.-Feb., 1963.

98) "Economics of Hydraulic-Turbine Selection"
R. E. Passmore - ASME-Paper 63-WA-128 for meeting Nov. 17-22, 1963.

99) "Modern Trends in Hydraulic Turbine Design in Europe"
G. A. Bovet - Trans. ASME, No. 75, August, 1953.

100) "Recent Developments in Francis Turbines."
Wm. J. Rheingans - Mechanical Engineering, Vol. 74, March, 1952.

101) "Nantahala Turbine"
J. P. Growdon, R. V. Terry, & H. H. Gnuse, Jr. - Trans. ASME, No. 68, October, 1964.

102) "Hydraulic Turbine Operation Difficulties"
J. Parmakian - Water Power Vol. 15, No. 10, October 1963.

103) "Hydraulic Turbine Deficiencies"
J. Parmakian - Water Power, July 1955.

104) "Operating Experience with High-Pressure Francis Spiral Turbines"
G. Gysi - Escher Wyss News Vol. 32, No. 2-3 1959.

105) "Hydraulic Installations"
P. Schmidheiny - Escher Wyss News Vol. 37, No. 1-2, 1964.

106) "Selecting Hydraulic Reaction Turbines"
R. E. Krueger - Engineering Monograph No. 20, Bureau of Reclamation, U. S. Department of the Interior - Denver, Colorado.

107) "Research on High Head Kaplan Turbine"
S. Saito - Tohoku University--Inst. High Speed Mechanics--Reports Vol. 15, 1963-1964.

108) "Recent Developments in Hydraulic Turbine Design and Application"
G. C. Hochwalt - ASCE Water Resources Eng. Conference - Preprint 141, March 8-12, 1965.

109) "Reduction of Fluid Friction Losses in Rotary Hydraulic Machines," R. S. Sproule, P. Koeller - Eng. Inst. Canada -- Trans. Paper ETC-64-Hydel, July 4, 1964.

110) "Hydraulic Turbines" - Three papers:
"Axial Turbines and Their Use in Modern Projects"
J. P. Blum (p. 1283-98).
"Tubular Turbines"
G. Hermann, Y. Van Pachterbeke (p. 1299-1318).
"Possibilities of Extending Usage of Hydraulic Turbines in Hydro-electric Developments"
R. de la Grand'rive, V. de Verdelhan (p. 1321-35)
Assn. des Ingenieurs Electriciens, Sortis de l'Institut Electrotechnique Montefiore - Bul. Vol. 75, No. 12, December 1962.

111) "Centrifugal Pumps Used as Hydraulic Turbines"
C. P. Kittredge - ASME Paper No. 59-A-136

IX. Drivers and Gears

112) "How to Select A. C. Motors for Economy with Capability"
C. R. Olson, E. S. McKelvy - Paper Trade Journal (p. 68 - 71), March 11, 1968.

113) "Prime Movers in Gearing"
V. W. Wigotsky, Design News, January 19, 1968

X. Textbooks

114) Principles of Turbomachinery
D. G. Shepherd, The Macmillan Company, New York.

115) Centrifugal and Axial Flow Pumps
A. J. Stepanoff John Wiley & Sons, Inc, New York.

116) Design and Performance of Centrifugal and Axial Flow Pumps and Compressors
André Kovats, Pergamon Press, The Macmillan Company, New York.

117) Marks' Mechanical Engineers' Handbook
Theodore Baumeister McGraw-Hill Book Company

118) Desalination by Reverse Osmosis
U. Merten, Massachusetts Institute of Technology Press

119) Electrical Engineering
W. H. Erickson, N. H. Bryant, John Wiley & Sons, Inc., New York

120) Corrosion Handbook
H. H. Uhlig, Behavior of Metals and Alloys in Sea Water.

APPENDIX B

This appendix includes the questionnaires which were sent to pump manufacturers and hydraulic turbine manufacturers.

Appendix B1 - Pump Questionnaire

Appendix B2 - Hydraulic Turbine Questionnaire

DYNATECH CORPORATION
Progress through Research 17 TUDOR STREET, CAMBRIDGE, MASS., 02139
617-UNiversity 8-8050

Appendix B1

Pump Questionnaire

Gentlemen:

Dynatech Corporation, an independent engineering research and development organization, is carrying out an analysis of pumping equipment for the Office of Saline Water, United States Department of the Interior. Our purpose is to determine the present state-of-the-art with respect to pumping systems suitable for use in reverse osmosis desalination plants. This information will help us to select the optimum pumping system arrangement for various sizes of desalination plants. The results of our study will be used by OSW in plant design work, preparation of economic analyses, and in allocating development funds to the pump industry.

We have already collected and analyzed standard catalog data from the principal manufacturers of pumps suitable for this type of desalination plant. Now we wish to review our preliminary conclusions with the manufacturers and to gather supplementary information. We are particularly interested in acquiring price and operating cost information for a number of specific pump requirements. With this letter, we are requesting your assistance to provide the information which we need on your products.

We have enclosed a brief note which outlines the objectives of our study, presents a summary of our project plan as a background for our requests, and then describes the specific information which we are requesting. We will be pleased to assist you in any way we can to gather this material.

Thank you very much for your assistance with this study.

Sincerely yours,

Kenneth E. Hickman, Manager
Fluid Mechanics Group

DYNATECH

Pumping System Information Request
O. S. W. Contract No. 14-01-0001-1462
"Evaluation of Existing and Unconventional Means
for Pumping and Energy Recovery in a
Reverse Osmosis Desalination Plant"

1. Project Objectives

We are carrying out an analysis of pumping equipment for the Office of Saline Water, United States Department of the Interior. The objectives of our study are as follows:

(1) To select the best types of pumping systems for use in reverse osmosis desalination plants
(2) To determine whether the pumps required are presently available and, if so, from whom
(3) To identify areas of pumping technology which may require OSW-funded development work by the pump manufacturers in order to meet the requirements of desalination plants

We request your assistance in providing information on your products which are suitable for this service. We would like to obtain both technical and cost information for particular pumps; perhaps a summary of our project plan will help to explain how we defined particular pump specifications for your consideration.

2. Project Summary

In the reverse osmosis process for water desalination, salt water pumped to pressures from 400 psi to 1500 psi flows past osmotic membranes. Pure water diffuses through the membranes while the remaining more-concentrated brine is rejected. The operating pressure level required for good membrane performance is set by the salinity of the input water; brackish water of low salinity (5,000 ppm) requires pressures of 400 to 800 psi while sea water (35,000 ppm) requires 1500 psi for economical desalination.

The desalination plant sizes considered in our study cover the range from 100,000 to 10 million gallons per day of fresh water output. The ratio of fresh water recovered to plant inlet flow is 40% for sea water plants and varies from 50% to 80% for brackish water plants. Therefore, the pumping systems to be considered cover the range of flow rates from 80 gpm to 19,000 gpm. The head requirements range from 800 ft to 3500 ft. Because of the high flow rate and high head requirements, the capital cost and operating costs of the pumping system are significant factors in the overall economy of the plant.

To meet our objective of selecting the best type of pumping system for each set of plant conditions, we plan to carry out an economic analysis of the capital and operating costs for a number of possible pumping system arrangements. The arrangements differ with respect to type of pump and manufacturer, pump speed, driver (electric motor, steam turbine, or diesel engine), and power recovery turbine hookup. To keep the effort required for the economic analysis within bounds, we have defined 30 discrete plant pump requirements which cover the complete range of flow rates and heads for the desalination plants. Figure 1 shows the 30 plant pump requirements and indicates their relationship to the plant operating conditions.

We have acquired general catalog information from the principal manufacturers of high-flow, high-head pumps and we have attempted to identify specific pumps in each manufacturer's line which can be matched to our 30 plant pump requirements. The next step in our program, the step which this letter initiates, is to review our selections with the manufacturers and to gather supplementary information. We want to obtain price and operating cost information for each of the 30 plant pump requirements, and for each speed for which pumps are available to meet a given plant.requirement. We also want to obtain information on pump characteristics, materials of construction, accessories and auxilary equipment, installation and maintenance requirements, and operating instructions.

We plan to use the cost data to develop graphical curves or equations which describe the relationship of cost to the pump performance requirements. Our work on turbines, electrical motors, and generators indicates that both the initial and operating costs of such equipment can be correlated effectively with horsepower for a given class of equipment. We expect that a similar type of correlation will be useful for the pump cost information. After these correlations are developed we will use computer techniques to compare costs of the various types of pumping systems which can be used to meet each plant pump requirement. These systems will vary with respect to the following elements:

type of pump and manufacturer
pump speed
type of driver
power recovery arrangement

The systems which yield the lowest total cost, amortized over a 30-year period, will be selected as the optimum pumping systems for the 30 selected plant requirements.

In addition to our recommendations on optimum pumping system design, we also will prepare for OSW a report which discusses a number of operating and design considerations which are pertinent to pumps for desalination service. These considerations include water pretreatment requirements, installation characteristics, pump material selection.

maintenance requirements, control systems, spare parts recommendations, and typical operating problems which may be encountered. The information to be included in this report will be sought from the manufacturers; supplementary information has already been gathered from the technical literature.

The final report of this project will be made available to the public by OSW. In this report, we will list the manufacturers who provided the information upon which the report is based. However, no identification of specific manufacturers will be made at any other point within the report; alternative pumps for specific plants will be identified by letter only (e.g., pump A, B, C). The cost and technical data provided by each manufacturer will be treated as confidential by Dynatech and OSW if so requested. Such confidential information will not be disclosed outside of these organizations.

3. Information Requested

The information which we request from you can be divided into two classes; information on specific pumps, and general operational and design information.

3.1 Specific Pumps

We would like to obtain performance and cost information for those pumps which you now manufacture which meet the flow and head requirements for individual plants taken from our list of 30 standard plants (Fig. 1). Enclosed with this letter is a list of your pumps which we have matched to some of the standard plants. Also enclosed is a set of one-page data sheets which indicate the information we would like to have for each pump, and which we hope will make your response easier to prepare.

Please review our list of standard plants to see if we have overlooked other pumps which you could offer, and whether we have selected the best one of your pumps for the plants we have already considered. Can the same plant requirements be met by pumps of different speeds? We will appreciate any additional information or comments which you may have on this pump selection phase of our study.

3.2 General Information

To assist us in preparing our report on operating and design considerations for desalination process pumps, we would like to obtain answers to a number of questions about your pumps. These questions are general in nature; the same answers may be applicable to a number of your pumps which were identified in the data sheets above. You may have already prepared brochures, reports, or operating manuals for your pumps which provide most of the answers we seek.

The questions which we have prepared are listed on several pages enclosed with this letter. We will contact you in about one week to determine the manner in which it would be most convenient for you to provide answers to us.

We would like to know of any pump development programs which are now in progress or are being planned at your company which would change the position of your company's products with respect to the reverse osmosis desalination process. This information will be treated as confidential by Dynatech and OSW if you so request.

Questions for Section 3.2: General Information - Pumping Systems

We would like to obtain general design and application information on those types of your pumps which meet the requirements for our standard plants. This note lists the information we need and the questions which we want to answer for each type of pump.

Your replies to some questions will be guided by the fact that the desalination plant is a utility. The pumps, together with their drivers and accessories, must operate continuously for long periods between scheduled shutdowns. Because of the nature of the process, the pumps will operate at constant speed with only small, slow variations of head and flow rate which occur as a result of long-term membrane performance fluctuations.

1. Pump Characteristics

   1.1 Efficient operating range: If possible, please provide typical head and efficiency vs. flow rate curves for appropriate pumps. Otherwise, please indicate the range of flow rates over which the efficiency remains within 4 points of its peak value.

   1.2 Physical Descriptions: Please provide a cross-section drawing of each type of pump or identify its type (e.g., vertical, horizontal, barrel, etc.). Please describe the following components and their materials of construction:

   inlet geometry: double or single suction
   impeller construction
   diffuser type
   thrust balancing features
   bearings
   seals recommended for our application, and their expected leakage characteristics; what alternative seal arrangements could be used, if any? What are the associated cost variations?

2. Drivers

2.1 Recommended drivers:

Do you recommend any particular drivers for these pumps?

What type of electrical motor is best for this application?

Are there any particular cautions you would advise with respect to the use of turbine or diesel engine drivers?

2.2 Packaged Systems

Do you offer driver-pump-accessories packages ready for installation as an option?

3. Accessories and Auxiliary Equipment

3.1 Spares

What spares should be kept on hand to assure rapid restoration of service at scheduled maintenance periods?

What spares are required to protect against failures?

3.2 Lube Oil System

Please provide a description and diagram of the lube oil system required for your pumps.

3.3 Please describe any external (to the pump) equipment required for the sealing systems you have recommended.

3.4 Controls

Please outline the controls required for safe and efficient operation of your pumps.

3.5 Couplings

Do you recommend any particular types of couplings for these pumps?

Does the type of driver to be used affect the choice of coupling type?

4. Installation Requirements

4.1 Inlet and Discharge Piping

Are there any special requirements with respect to inlet and discharge piping arrangement which should be observed for best performance with your pumps?

4.2 Foundation Requirements

Please describe the foundation requirements for your pumps.

4.3 Set-up Procedures

Please outline any special procedures or precautions to be observed when setting up your pumps in a new plant

4.4 General Arrangement

If possible, please furnish diagrams showing a typical installation of your pumps and their relation to the driver and to auxiliary systems.

5. Maintenance

Please describe the maintenance requirements for your pumps.

What program for maintenance and inspection operations do you recommend?

What is the life expectancy for each of the following pump components in saline water service:

seals
rotors
diffusers
bearings
casings

What are the types of failures which occur in these pumps?

6. Operating Instructions

Please provide typical operating manuals if possible.

What are the procedures to be followed to start up and shut down your pumps?

What are the types of operating problems encountered in service and how may they be avoided?

7. Costs

7.1 Pump Prices

The information you have provided on specific pumps includes price information. Do these prices include any accessories, spares, or auxiliaries?

Do you have any correlations on pump price vs. another parameter (e.g., horsepower, head, impeller diameter, speed, flow rate, or number of stages) which we could use to guide our economic comparisons?

7.2 For the pumps you have described in response to our particular plant requirements, please indicate costs where possible for the following items:

maintenance
spares
foundation and installation costs
auxiliaries: lube oil, seal, and control systems
packaged pump-driver-auxiliary systems where available
operating costs exclusive of driver power costs, if any

BY JPT DATE 10/6/67 SUBJECT PROJECT SAL-5-1 SHEET NO. ____ OF ____

CHKD. BY ____ DATE ____ DIFFERENT PLANT SIZES JOB NO. ____

HEAD & CAPACITIES

867 ft	#1 86.75 gpm	#4 867.5 gpm	#7 8,675 gpm
867 ft	#2 99.30 gpm	#5 993 gpm	#8 9,930 gpm
867 ft	#3 138.80 gpm	#6 1,388 gpm	#9 13,880 gpm
1320 ft	#10 86.75 gpm	#13 867.5 gpm	#16 8,675 gpm
1320 ft	#11 99 30 gpm	#14 993 gpm	#17 9,930 gpm
1320 ft	#12 138.80 gpm	#15 1,388 gpm	#18 13,880 gpm
1765 ft	#19 86.75 gpm	#22 867.5 gpm	#25 8,675 gpm
1765 ft	#20 99.30 gpm	#23 993 gpm	#26 9,930 gpm
1765 ft	#21 138.80 gpm	#24 1,388 gpm	#27 13,880 gpm
3340 ft	#28 173.3 gpm	#29 1,733 gpm	#30 17,330 gpm

Pump No. System #

I. OPERATING CONDITIONS

Suction Temperature	70 °F
Suction Pressure	1 atm.
Density	lbm/ft³
Viscosity	1 centipoise
Particle size after prefiltering	5 μ
Acidity level	pH = 6 to 8
Saline Water type	
ppm Salt.	
Head developed	feet
Flow delivered	gpm

II. PUMP CHARACTERISTICS.

Head-Flow Curve (attach if possible).	
NPSH	feet
Horsepower (BHP)	HP
Efficiency	%
Starting Torque	
Full-load Torque	
Shaft speed	rpm
Self-priming or not	

III. PHYSICAL DESCRIPTION.

Type	
Over-all Dimensions	
Impeller diameter	
Weight	
Inlet port Dimensions	
Discharge port Dimensions	

IV. MATERIALS RECOMMENDED

V. COSTS (WITH SELECTED MATERIALS).

VI. REMARKS.

Progress through Research DYNATECH CORPORATION
17 TUDOR STREET, CAMBRIDGE, MASS., 02139

617-UNiversity 8-8050

Appendix B2

Hydraulic Turbine Questionnaire

Gentlemen:

Dynatech Corporation, an independent engineering research and development organization, is carrying out an analysis of hydraulic power recovery equipment for the Office of Saline Water, U.S. Department of the Interior. Our purpose is to determine the present state of the art with respect to power recovery systems suitable for use in reverse osmosis desalination plants. This information will help us to select the optimum power recovery system for various sizes of desalination plants. The results of our study will be used by OSW in plant design work, preparation of economic analyses, and in allocating development funds to the hydraulic turbine industry.

We have already collected and analyzed general information from literature made available to us by major hydraulic turbine manufacturers. Now we wish to complete our preliminary information and acquire price and operating cost information for a number of specific hydraulic turbine requirements. With this letter, we are requesting your assistance to provide this supplementary information.

We understand that hydraulic turbines are not "off-the-shelf" equipment, but machines "tailored" to each customer's needs. We have enclosed a brief note which outlines the objectives of our study, presents a summary of our project plan as a background for our requests and then describes the specific information which we are requesting. We will be pleased to assist you in any way we can to gather this material.

Thank you very much for your assistance with this study.

Sincerely yours,

Kenneth E. Hickman, Manager
Fluid Mechanics Group

KEH:da

Enclosure

Power Recovery System Request
O. S. W. Contract No. 14-01-0001-1462
"Evaluation of Existing and Unconventional Means
for Pumping and Energy Recovery in a
Reverse Osmosis Desalination Plant"

1. Project Objectives

We are carrying out an analysis of pumping and power recovery equipment for the Office of Saline Water, United States Department of the Interior. The objectives of our study are:

(1) To select the best types of pumping systems for use in reverse osmosis desalination plants
(2) To select the best types of power recovery systems for use in reverse osmosis desalination plants.
(3) To study the integration of these power recovery systems into the pumping system and to determine the resulting economic advantages.

We request your assistance in providing information on your products which are suitable for this service. We would like to obtain both technical and cost information for particular power recovery systems; perhaps a summary of our project plan will help to explain how we defined particular hydraulic turbine specifications for your consideration.

2. Project Summary

In the reverse osmosis process for water desalination, salt water pumped to pressures from 400 psi to 1500 psi flows past osmotic membranes. Pure water diffuses through the membranes while the remaining more concentrated brine is rejected. The operating pressure level required for good membrane performance is set by the salinity of the input water; brackish water of low salinity (5,000 ppm) requires pressures of 400 to 800 psi while sea water (35,000 ppm) requires 1500 psi for economical desalination.

The desalination plant sizes considered in our study cover the range from 100,000 to 10 million gallons per day of fresh water output. The ratio of fresh water recovered to plant inlet flow is 40% for sea water plants and varies from 50% to 80% for brackish water plants. Therefore, the pumping systems to be considered cover the range of flow rates from 80 gpm to 19,800 gpm. The head requirements range from 800 ft to 3500 ft. Because of the high flow rate and high head requirements, the capital cost and operating costs of the pumping system are significant factors in the overall economy of the plant.

The concentrated brine which remains behind as the fresh water passes through the membranes is discharged from the membrane assembly at high pressure. The power associated with this pressure can be recovered by a hydraulic turbine and used to drive the main system pump or a generator. Because of the high flow-rate and high head available in the concentrated brine, the power recovery obtained with a hydraulic turbine can be a significant factor in the overall economy of the plant.

To keep the effort required for the economic analysis within bounds, we have defined 30 discrete plant pump requirements which cover the complete range of flow rates and heads for the desalination plants. We then assumed several possible values for the pressure drop on the brine side of the membranes. The resulting head values combined with the brine discharge flow rates define a number of hydraulic turbine specifications.

Figure 1 shows the 30 plant pump requirements and indicates their relationship to the plant operating conditions. Figure 2 shows the flows and heads available for power recovery, together with horsepower values corresponding to reasonable turbine efficiencies. Figure 3 shows a general plant layout.

We would like to obtain price and operating cost information for each of the hydraulic turbine requirements which are circled in red on Figure 2. We also want to obtain more information on hydraulic turbine characteristics, materials of construction, accessories, controls and auxiliary equipment, installation and maintenance requirements, and operating instructions.

The information for the requirements circled on Figure 2 will be used to establish graphical curves or equations which describe the relationship of cost to turbine performance over the whole range of our requirements. Our work on steam turbines, electrical motors, and electrical generators indicates that both the initial and operating costs of such equipment can be correlated effectively with horsepower for a given class of equipment. We expect that a similar type of correlation will be useful for the hydraulic turbine cost information. These correlations will be included in a general computer program which will establish the economic savings achieved with a power recovery system in relation to desalination plant size. The systems evaluated by computer techniques will vary with respect to the following elements.

- type of pump and manufacturer
- pump speed
- type of driver
- hydraulic turbine efficiency
- hydraulic turbine speed
- power recovery arrangement (pump vs. generator drive)

The systems which yield the lowest total cost, amortized over a 30-year period, will be selected as the optimum pumping and energy recovery system for the 30 selected plant requirements.

In addition to our recommendations on the feasibility of power recovery systems, we also will prepare for OSW a report which discusses a number of operating and design considerations which are pertinent to hydraulic turbines for desalination service. These considerations include water condition, installation characteristics, material selection, maintenance requirements, control systems, spare parts recommendations, and typical operating problems which may be encountered. The information to be included in this report will be sought from the manufacturers; supplementary information has already been gathered from the technical literature and textbooks describing the subject.

The final report of this project will be made available to the public by OSW. In this report, we will list the manufacturers who provided the information upon which the report is based. However, no identification of specific manufacturers will be made at any other point within the report; alternative hydraulic turbines for specific plants will be identified by letter only (e.g., hydraulic turbine A, B, C). The cost and technical data provided by each manufacturer will be treated as confidential by Dynatech and OSW if so requested. Such confidential information will not be disclosed outside of these organizations.

3. Information Requested

The information which we request from you can be divided into two classes, information on specific hydraulic turbines, and general operational and design information.

3.1 Specific Hydraulic Turbines

We would like to obtain performance and cost information on those hydraulic turbine applications which have been circled in red on Figure 2. These applications have been selected in order to include a range of typical requirements:

- low flow, low head, low horsepower
- low flow, high head, intermediate horsepower
- medium flow, high head, intermediate horsepower
- high flow, low head, intermediate horsepower
- high flow, high head, high horsepower

The head and flow values are given as operating conditions for a particular turbine. The horsepower given here has been calculated by assuming an efficiency of 85%. Four synchronous speeds are being considered for these machines.

- 1800 rpm
- 1200 rpm
- 900 rpm
- 3600 rpm

Enclosed with this letter is a set of one-page data sheets which indicate the information we would like to obtain for each turbine and which we hope will make your response easier to prepare. We will appreciate any additional information or comments which you may have on this hydraulic turbine selection phase of our study.

3.2 General information

To assist us in preparing our report on operating and design considerations for power recovery systems, we would like to obtain answers to a number of questions about your hydraulic turbines. These questions are general in nature. You may have already prepared brochures, reports, or operating manuals for your hydraulic turbines which provide most of the answers we seek.

The questions which we have prepared are listed on several pages enclosed with this letter. We will contact you in about two weeks to determine the manner in which it would be most convenient for you to provide answers to us.

We would like to know of any hydraulic turbine development programs which are now in progress or are being planned at your company.which would change the position of your company's products with respect to reverse osmosis process. This information will be treated as confidential by Dynatech and OSW if you so request.

QUESTIONS: GENERAL INFORMATION - HYDRAULIC TURBINES

We would like to obtain general design and application information on those types of your hydraulic turbines which meet the requirements for our standard power recovery systems. A general layout of the plant has been attached to this questionnaire in order to indicate conditions of installation and operation.

Your replies to some questions will be guided by the fact that the desalination plant is a utility. The hydraulic turbines, either directly connected to the pump by means of gears, or driving an electrical generator, should operate continuously for long periods between shutdowns. Because of the nature of the process, the hydraulic turbines will operate at constant speed with only small, slow variations of head and flow-rate which occur as a result of long-term membrane degradation.

1. Hydraulic Turbine Characteristics

1.1 Physical Descriptions

Please provide information as follows for your hydraulic turbines:
Type: Francis or Pelton
How many jets per runner (Pelton)?
Guide vanes (size, number) for Francis?
Runner construction (material)
Casing type and construction (material)
Injector size and geometry
Inlet guard valves (type, recommended size and location)
Is fixed gate type injection used? (for Pelton turbines)
Draft tube geometry (if any)
Thrust balancing features
Bearing types
Seals or shaft packings recommended for our application, and their expected leakage characteristics; what alternative seal arrangements could be used, if any?

What are the associated cost variations?

Would a leakage pump be needed (Francis)?

Please provide a cross-section drawing of each type of turbine if possible.

1.2 Efficient operating range: if possible, please provide typical hp and efficiency vs. flow rate curves for appropriate hydraulic turbines. Otherwise, please indicate the range of flow rates over which the efficiency remains within 4 points of its peak value.

2. Governors

Would you recommend a governor for this type of operation?

Would it be best to have an oil pressure governor or an electric governor?

What would be the size and cost of the governor including its controls?

3. Hydraulic Turbine Drive Arrangement

What type of generator is best for this application? (Voltage and speed)

Are there any particular cautions you would advise with respect to the use of a hydraulic turbine as a supplementary source of power when the primary source is an electric motor, steam turbine or diesel engine?

Do you offer hydraulic turbine-generator packages ready for installation as an option?

4. Accessories and Auxiliary Equipment

4.1 Spares

What spares should be kept on hand to assure rapid restoration of service at scheduled maintenance periods?

What spares are required to protect against failures?

4.2 Lube oil system

Please provide a description and diagram of the lube oil system required for your hydraulic turbines.

4.3 Sealing Systems

Please describe any external equipment (to the hydraulic turbine) required for the sealing systems you have recommended.

4.4 Controls

Please outline the controls required for safe and efficient operation of your hydraulic turbines (hand operated or automatic)

4.5 Couplings

Do you recommend any particular types of couplings for these units? Will the selection of the power recovery system (generator or direct pump drive) affect the choice of coupling type?

5. Installation Requirements

5.1 Inlet and Discharge Piping

Are there any special requirements with respect to inlet piping arrangement which should be observed for best performance of your hydraulic turbines?

Could your hydraulic turbines directly discharge into a sump? If not, would you please describe discharge piping arrangement, or any draft tube requirements?

5.2 Foundation requirements

Please describe the foundation requirements for your hydraulic turbines.

5.3 Set-up Procedures

Please outline any special procedures or precautions to be observed when setting up your hydraulic turbines in a new plant.

5.4 General Arrangement

If possible, please furnish diagrams showing a typical installation of your hydraulic turbines and their relation to the generator. This will help us in modifying the general plant layout accordingly to our power recovery system design.

6. Maintenance

Please describe the maintenance requirements for your hydraulic turbines.

What program for maintenance and inspection operations do you recommend?

What is the life expectancy for each of the following hydraulic turbine components in saline water service:

- seals or packings
- casings or housings
- needle tip
- injector throat ring
- bucket splitters
- deflector
- valves
- runner bands (Francis)
- guide vanes (Francis)
- stationary rings (Francis)
- distributor lining (Francis)
- runner blading (Francis)

How are wearing symptoms detected? What is their effect on the efficient operation of the machines?

7. Operating instructions

Please provide typical operating manuals if possible.

What are the procedures to be followed to start up and shut down your units?

What precautions are needed in the case of pressure surge?

What are the types of operating problems encountered in service and how may they be avoided?

8. Costs

8.1 Hydraulic Turbine Prices

The information you have provided on your hydraulic turbines includes their cost. Do these prices include any accessories, valve requirements, spares or auxiliaries?

Do you have any correlations on hydraulic turbine price versus another parameter (e.g., horsepower output, head, runner diameter, specific speed, specific discharge) which we could use to guide our economic comparisons?

8.2 For the hydraulic turbines you have described in response to our particular plant requirements, please indicate costs whenever possible for the following items:

- maintenance
- spares
- foundation & installation costs
- auxiliaries: lube oil, seals or packings, bearings, regulation and control systems, governor.
- packaged hydraulic turbine - generator systems where available
- operating costs exclusive of recovered power savings.

Pump Head	% Rec.	% Head Loss	10^5 GPD 0	5	10	15	10^6 GPD 0	5	10	15	10^7 GPD 0	5	10	15
867 FT.	50	FT	867	822	777	732	867	822	777	732	867	822	777	732
		GPM	17.3			→	173.			→	1730			→
		HP	3.2	3.05	2.9	2.7	32.	30.5	29.	27	320	305	290	270
	70	FT	867	822	777	732	867	822	777	732	867	822	777	732
		GPM	29.9			→	299.			→	2990			→
		HP	5.5	5.3	4.9	4.7	55.	53.	49.	47.	550	530	490	470
	80	FT	867	822	777	732	867	822	777	732	867	822	777	732
		GPM	69.4			→	694			→	6940			→
		HP	12.9	12.2	11.5	10.9	129	122	115	109	1290	1220	1150	1090
1320 FT.	50	FT	1320	1250	1180	1115	1320	1250	1180	1115	1320	1250	1180	1115
		GPM	17.3			→	173			→	1730			→
		HP	4.9	4.6	4.4	4.1	49.	46.	44	41	490	460	440	410
	70	FT	1320	1250	1180	1115	1320	1250	1180	1115	1320	1250	1180	1115
		GPM	29.9			→	299			→	2990			→
		HP	8.5	8.0	7.6	7.1	85.	80	76	71	850	800	760	710
	80	FT	1320	1250	1180	1115	1320	1250	1180	1115	1320	1250	1180	1115
		GPM	69.4	–		=	694.	-		=	6940	–		=
		HP	19.6	18.6	17.6	16.6	196.	186	176	166	1960	1860	1760	1660
1765 FT.	50	FT	1765	1675	1590	1495	1765	1675	1590	1495	1765	1675	1590	1495
		GPM	17.3			→	173.			→	1730			→
		HP	6.5	6.2	5.9	5.5	65.	62.	59.	55.	650	620	590	550
	70	FT	1765	1675	1590	1495	1765	1675	1590	1495	1765	1675	1590	1495
		GPM	29.9			→	299.			→	2990			→
		HP	11.3	10.7	10.2	9.6	113.	107.	102.	96.	1130	1070	1020	960
	80	FT	1765	1675	1590	1495	1765	1675	1590	1495	1765	1675	1590	1495
		GPM	69.4			→	694.			→	6940			→
		HP	26.3	24.9	23.7	22.3	263	249	237	223	2630	2490	2370	2230
3340 FT.	40	FT	3340	3175	3000	2835	3340	3175	3000	2835	3340	3175	3000	2835
		GPM	104.			→	1040			→	10,400			→
		HP	74.5	70.9	67.	63.2	745	709	670	632	7450	7090	6700	6320

Figure 2 - Hydraulic Turbine Design Requirements

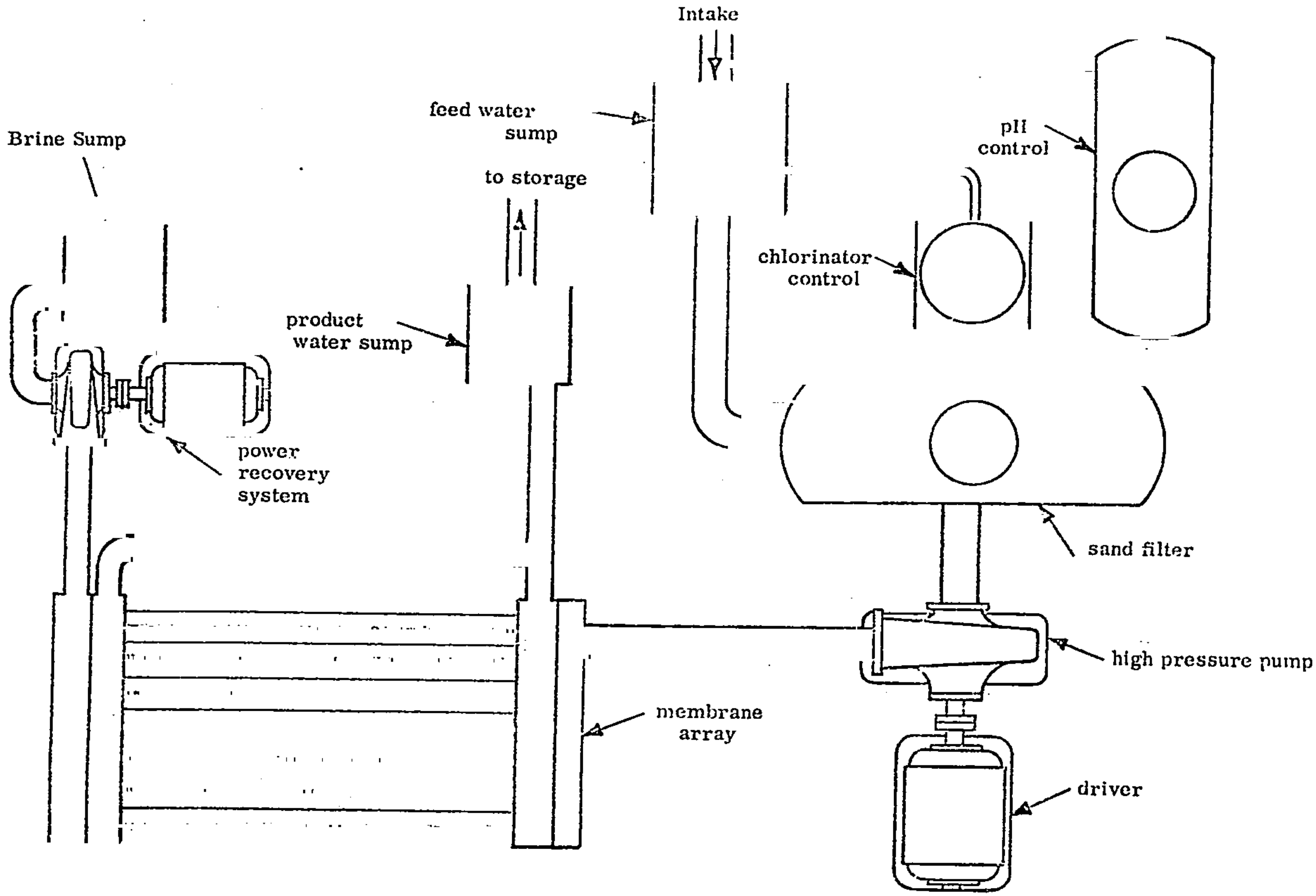


Figure 3 - General Plant Layout

Pump Speed N

Head (psia)	Fresh Water Output (GPD)	Head-Capacity Requirement	%R Factor	Q gpm	HP Rating	Plant Pump Requirement
400 (867 ft)	10^5	No. 1	80	86.75	25.3	No. 1
			70	99.30	28.9	2
			50	138.80	40.4	3
	10^6	2	80	867.50	253	4
			70	993	289	5
			50	1,388	404	6
	10^7	3	80	8,675	2,530	7
			70	9,930	2,890	8
			50	13,880	4,040	9
600 (1320 ft)	10^5	4	80	86.75	38.6	10
			70	99.30	44	11
			50	138.80	61.5	12
	10^6	5	80	867.50	386	13
			70	993	440	14
			50	1,388	615	15
	10^7	6	80	8,675	3,860	16
			70	9,930	4,400	17
			50	13,880	6,150	18
800 (1765 ft)	10^5	7	80	86.75	51.6	19
			70	99.30	58.9	20
			50	138.80	82.5	21
	10^6	8	80	867.50	516	22
			70	993	589	23
			50	1,388	823	24
	10^7	9	80	8,675	5,160	25
			70	9,930	5,890	26
			50	13,880	8,230	27
1500 (3540 ft)	10^5	10	40	173.30	195	28
	10^6	11	40	1,733	1,950	29
	10^7	12	40	17,330	19,500	30

Figure 1 - Standard plant pump requirements

HYDRAULIC TURBINE

IV - Physical Description.

Type		
Number of jets		
Guide vanes mean diameter		
Casing or Housing dimensions		
Wheel mean diameter		
Runner outlet diameter		
Jet diameter		
Needle size		
Inlet valves		
Discharge valves		
Relief valves		
Inlet Piping dimensions		
Discharge Piping dimensions		

V - Materials Recommended.

VI - Costs (with selected materials)

Hydraulic Turbine & Controls	
Governor.	

VII Accessories and Spares.

(1)

HYDRAULIC TURBINE

I - Operating Conditions.

Suction Temperature
Suction Pressure
Discharge Temperature
Discharge Pressure
Head Available
Flow Delivered
Density
Viscosity
Particle Size (after prefiltering)
Acidity Level
Saline Water Type
ppm Salt (salinity)

II - Hydraulic Turbine Characteristics.

Characteristic Curves (attach if possible)
Horsepower Output (BHP)
Efficiency (peak)
Shaft Speed.

III - Comments.

*U. S. GOVERNMENT PRINTING OFFICE : 1969 O - 356-580